MAY 2 1930

Heating · Piping and Air Conditioning

May, 1930

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By SABIN CROCKER, SAMUEL R. LEWIS *and* PLATTE OVERTON

Piping—
By C. T. BAKER, A. A. SELLKE *and* G. D. WINANS

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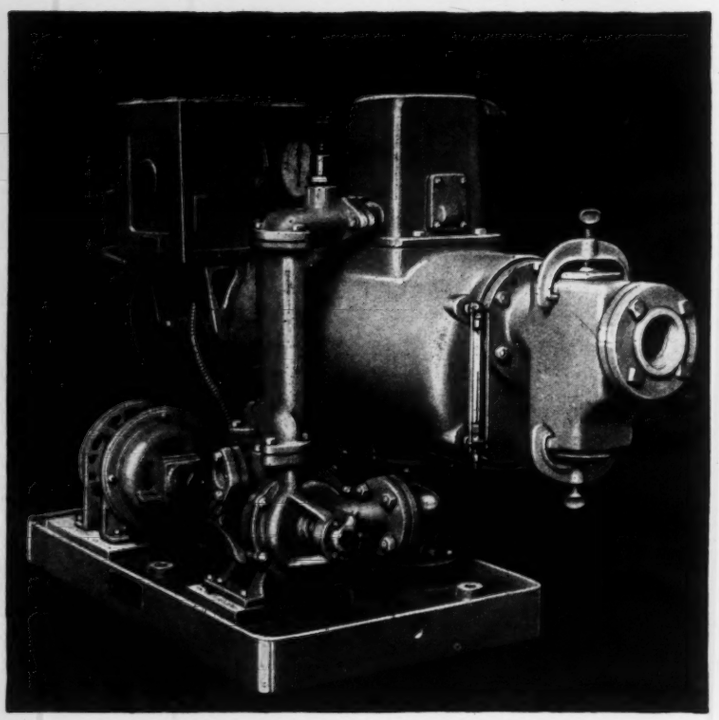

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Heating · Piping and Air Conditioning

Vol. II May, 1930 No. 5

Heating and Ventilating Delray Power House 3*

By Sabin Crocker

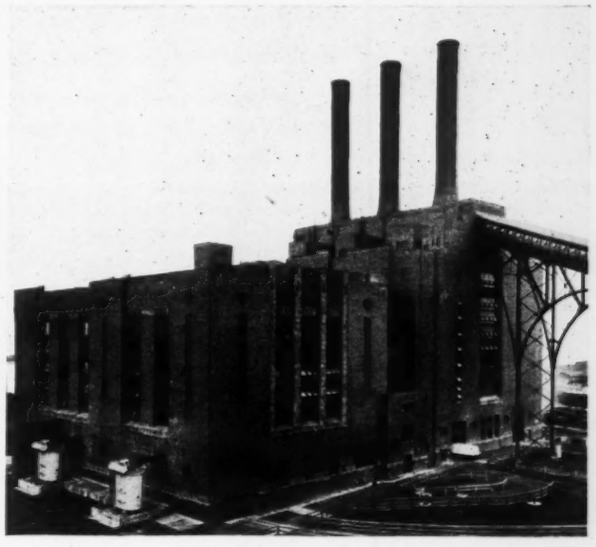
SPECIAL problems are encountered in power plant heating and ventilating design which require different treatment from other buildings. In winter, heating and ventilating a power plant involves providing heat when and where required, and sweeping out escaped steam which otherwise would condense on the ceiling and windows. In summer the problem concerns circulating fresh air so as to remove overheated air from portions of the building not adequately cooled by natural circulation.

The presence of both hot and cold equipment, and the fact that the various units of this equipment may be in or out of service irrespective of weather conditions, requires a certain background of past experience on which to base design. Usual methods of computing required radiation based on heat loss through walls and roof and on expected air changes have little significance under these circumstances.

In attempting to explain the design and operation of a power house heating and ventilating system, it is convenient to treat separately the two extreme conditions encountered, which will be designated here as summer conditions and winter conditions. Obviously, many operating combinations will be used to suit intermediate conditions of weather or escaped steam, which need not be discussed in detail. It is convenient to explain the simplest case first; i. e., summer conditions.

Summer Conditions

Summer conditions for ventilating the entire power house are shown in Fig. 1. The main turbine room is cooled by opening ventilating sash in the large windows on three sides of the room. When this does not afford sufficient cooling effect by natural circulation, heated air in the upper part of the room is drawn off through a duct running lengthwise of



the turbine room in a monitor on the roof. This in turn causes more fresh air to flow in through the windows, thus sweeping out overheated air in the upper part of the room. The duct on the roof leads to exhauster fans located in the auxiliary bay fan room, which discharge to outdoors in summer or to the boiler house in winter. Air enters the roof duct through an adjustable slot which is made in sections so that the ventilating effect produced in different parts of the room is subject to control. The capacity of the exhauster fans is sufficient to provide five air changes per hour for the main turbine room in addition to thirty changes in the auxiliary turbine room.

The auxiliary turbine room (located in the auxiliary bay between the main turbine room and boiler house, Fig. 1) is cooled by a set of fans which force in cool air through registers located in the side wall of the room. Another set of fans forces filtered fresh air through ducts leading to the generators for cooling them to remove heat developed by electrical losses in their windings. This air escapes into the auxiliary turbine room and then is removed by

* This article was written from notes of an illustrated talk given at the February 17, 1930, meeting of the Michigan Chapter A. S. H. V. E.

the large exhauster fans along with air which entered through the wall registers. The capacity of the wall register sets is sufficient to provide five air changes per hour in the auxiliary turbine room, and that of the generator cooling sets twenty-five air changes, making a total of thirty air changes per hour which can be forced into this room.

As shown in Fig. 1, the same set of four 100,000 c.f.m. exhauster fans serves to remove overheated air from both the main and auxiliary turbine rooms. Air can enter the ventilating fans in the auxiliary bay fan room through windows in the ends of the fan room or through louvres in a penthouse on the roof (see small insert of auxiliary bay in upper left hand corner of Fig. 2). Air for cooling the auxiliary generators is drawn into the fans through filters of the surface type moistened with oil. This precaution is necessary to avoid having dust collect in the generator windings.[1] Forced ventilation of the auxiliary turbine room in summer is essential, since it is located in a long narrow pocket having windows in the extreme ends only.

In ventilating the boiler house, use is made of air intended for combustion while on its way to the forced draft fans supplying the boilers. Fresh air enters the boiler house through ventilating sash which are left open at strategic points, sweeping overheated air ahead of it into the suction of the forced draft fans (See Fig. 1). The desired cooling effect can be obtained at any point by choosing the location of windows opened. Drafts are avoided in cool weather by taking air for combustion in through roof monitors and openings in the uptake housings.

Winter Conditions

Winter operation of the ventilating system in conjunction with the direct heating system is shown in Fig. 2. In winter, the ventilating system is used (a) as a means of sweeping out moisture-laden air to prevent condensation on the turbine room ceiling and windows, and (b) to furnish tempered air to the auxiliary turbines and the auxiliary turbine room.

Actual heating of the building is done with direct radiation placed around the exposed walls of turbine room, condenser room, and boiler house mezzanine and operating floors. The amount of direct radiation installed is based on past experience in similar plants rather than on computed heat losses.

Methods used for computing heat loss from ordinary buildings become too involved when applied to a power plant. Steam and electric machinery in the plant gives off a large amount of heat, but the electrical load and number of units in service varies and no definite amount of heat can be counted on when needed. The problem is further complicated by the presence of cold surfaces on equipment such as condensers, circulating pumps and connecting piping which may contain water at an entering temperature as low as 33 deg. F.

Another hindrance to basing installed radiation on

[1] The main generators are cooled by a closed system in which the same air is used over and over and cooled by circulating water in a fin-tube heat exchanger known as a generator air cooler.

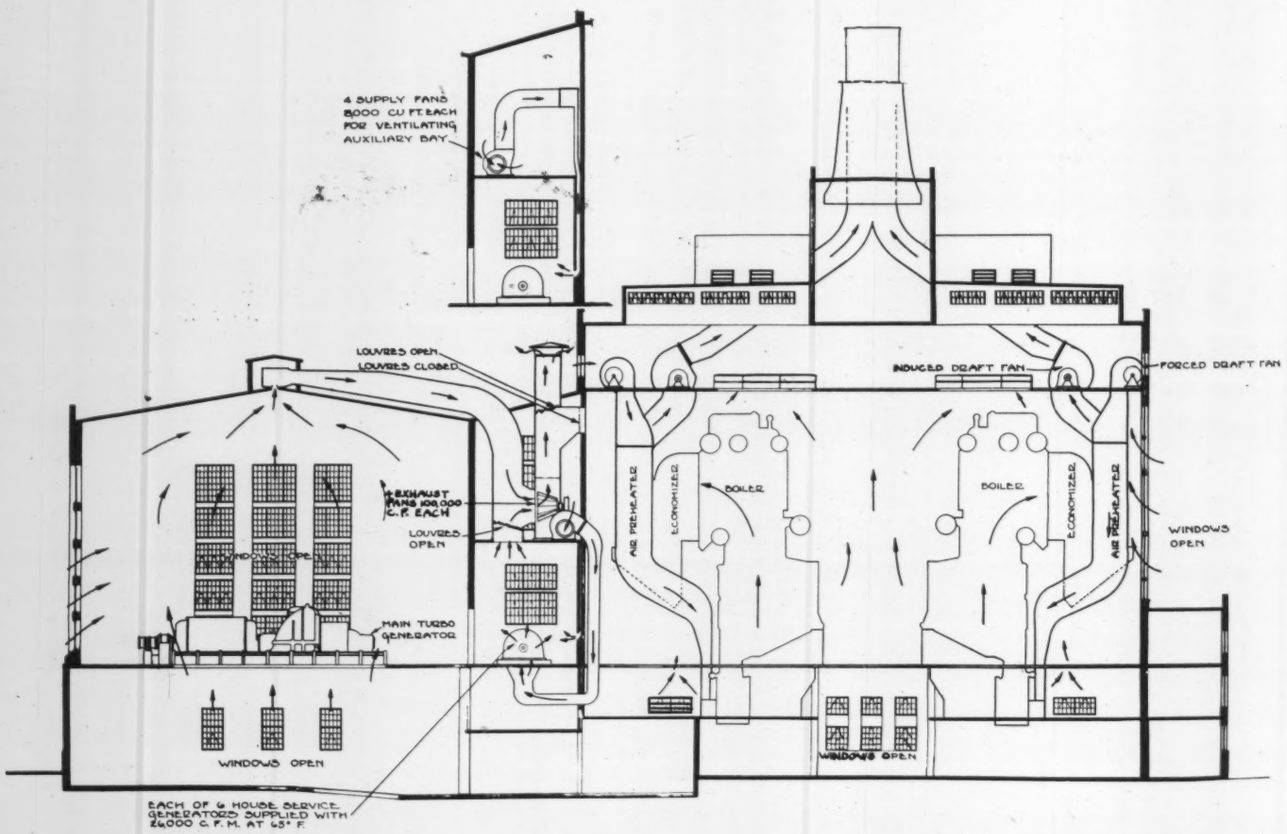


FIG. 1—OPERATION OF THE VENTILATING SYSTEM UNDER SUMMER CONDITIONS. AIR QUANTITIES MEASURED AT 70 F

computed heat loss is the unbroken height of turbine room and boiler room. In both of these rooms the operating floors are located at the lower levels so that any heat supplied there tends to rise and shirk doing its full duty.

For these reasons the problem mainly is one of installing sufficient radiation near the operators to give them a satisfactory sensation of radiated warmth, irrespective of the average room temperature, at the same time offsetting objectionable down drafts and the proximity of cold equipment.

Direct radiation has been used for this purpose, both because of a lack of concentration of the heating load, and because of the localized results wanted. It is possible, however, that unit heaters of the suspended type could be used to advantage in certain locations in place of some of the direct radiation. Such heaters would be a convenient means of eliminating sweating on the cold surfaces of condensers and their circulating pumps, which causes some annoyance with the present arrangement.

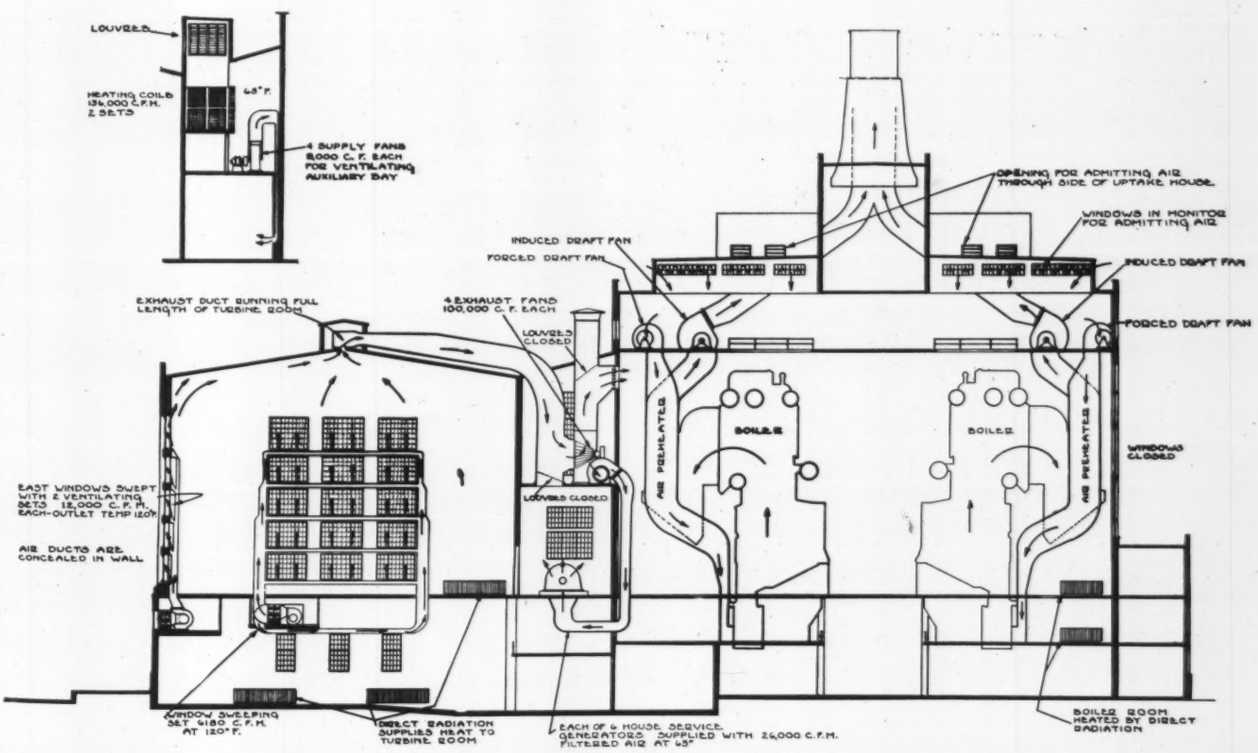
The amount of radiation installed and its location is determined with respect to the amount used in previous plants in these locations, taking into account the relative amount of exposed wall surface and the cubic contents involved. Experience has shown that considerable radiation is required around the boiler room operating floor and in the condenser room throughout the heating season. The direct radiation on the turbine room floor is not used in ordinary winter weather, except when the turbines are out of commission and the room would otherwise cool down. Direct radiation, or unit heaters, are required, of course, in offices, toilets and locker rooms, isolated shops and the like.

Window Sweeping System

Ventilation in winter is required in two principal parts of the building; viz., the main turbine room and the auxiliary turbine room. In the main turbine room, the ventilating system for the ultimate plant will consist of four window sweeping sets arranged to draw in fresh air from outdoors through heaters and discharge this air, after heating to 120 F, through ducts leading to slots just inside and below the turbine room windows. This provides a curtain of heated air for each window, thus preventing condensation and frosting on the windows and at the same time furnishing a supply of air to sweep the turbine room ceiling and prevent sweating there. The window sweeping sets have sufficient capacity to furnish slightly less than one air change per hour in the main turbine room.

The moist air is removed in turn through a duct in the monitor on the turbine room roof which leads to the large exhauster fans in the auxiliary bay fan room. In winter this warm and more or less moist air is discharged through a duct leading to the upper part of the boiler house, where it is picked up by the forced draft fans.

While it is not necessary to use the turbine room ventilating system at all times in winter, experience has shown it to be desirable equipment for use when and as required for eliminating excess moisture which has escaped through damaged shaft packing, valve stuffing boxes and the like. Recent improve-


LOUVRES
HEATING COILS 136,000 C.F.M. 2 SETS
60° F.
4 SUPPLY FANS 80,000 C. F. EACH FOR VENTILATING AUXILIARY BAY
OPENING FOR ADMITTING AIR THROUGH SIDE OF UPTAKE HOUSE
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EXHAUST DUCT RUNNING FULL LENGTH OF TURBINE ROOM
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BOILERS
AIR PREHEATER
WINDOWS CLOSED
EAST WINDOWS SWEPT WITH 2 VENTILATING SETS 12,000 C.F.M. EACH-OUTLET TEMP 120°
AIR DUCTS ARE CONCEALED IN WALL
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DIRECT RADIATION SUPPLIES HEAT TO TURBINE ROOM
EACH OF 4 HOUSE SERVICE GENERATORS SUPPLIED WITH 24,000 C.F.M. FILTERED AIR AT 45°
BOILER ROOM HEATED BY DIRECT RADIATION

FIG. 2—OPERATION OF THE VENTILATING SYSTEM UNDER WINTER CONDITIONS. AIR QUANTITIES MEASURED AT 70 F

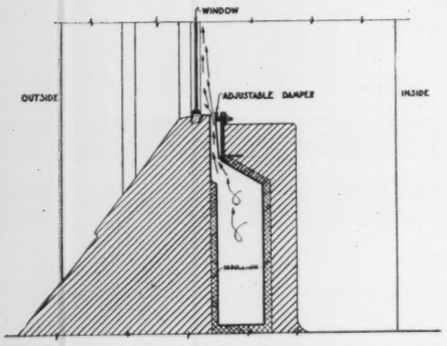


FIG. 3—DETAIL OF AIR DUCT AND OUTLET AT WINDOW

ments in shaft packing design have greatly reduced the quantity of sealing steam escaping into a turbine room under normal operating conditions, and as a result the need for a turbine room ventilating system in a plant built today is not as imperative as in plants built 6 or 8 years ago.

In the case of Delray P. H. 3, it was decided that the extra expense of such a ventilating system was warranted as insurance against the decidedly objectionable conditions which might come about through a temporary necessity for operating turbines with their shaft packings in bad condition. Under normal conditions such leakage can be kept down to an amount which practically is unnoticeable, but in case of damage to one or more sets of shaft packing enough steam can escape to cause a great deal of annoyance if allowed to condense on the windows and ceiling.

Other Methods of Preventing Sweating Proved Unsuccessful

Numerous other means of preventing sweating on the windows and ceiling have been tried in lieu of a ventilating system, but with indifferent success when bad moisture conditions were encountered. In one plant, a monitor with ventilating sash was installed along the ridge of the turbine room roof, providing a natural vent for the warm, moisture-laden air. The result was, however, that the moment the interior atmosphere came in contact with the cold surfaces of the monitor, serious condensation took place. In another instance, double sash were used to prevent fogging, but the results were not satisfactory. Insulating the turbine room roof was tried at one plant with a view to keeping the ceiling temperature above the dew point of the inside air, but with only partial success. To have installed sufficient insulation on a roof of this size would have cost more than the alternate solution offered in a ventilating system.

Air for Window Sweeping

A detail of the duct work and adjustable slot used to regulate the distribution of air for the window sweeping system is shown in Fig. 3. A duct of this kind is installed in the sill below each turbine room window and an equivalent duct with an ornamental panel front made of steel is provided below each tier of upper sash in the window. All duct-work above the turbine room floor is hidden in the walls or concealed by panelling for the sake of appearance.

Consideration was given also to making provision for possible future ventilating sets and duct-work for sweeping the turbine room ceiling. This system would serve as an adjunct to the present window sweeping sets by discharging air at 120 F across the ceiling from ducts running lengthwise of the turbine room in the upper corners just below the roof. This air would be exhausted through the duct in the roof monitor, along with that entering the turbine room through the window sweeping system. The air handling capacity of the roof sweeping system would approximate one air change per hour for the cubic contents of the turbine room. It is not expected that this ventilating system will be required, but space has been reserved for it in the construction of the building so that it can be added at some future time if and when found necessary.

Auxiliary Turbine Room Ventilation

Winter ventilation of the auxiliary turbine room utilizes the same equipment as summer ventilation with the addition of tempering heaters which can be used as occasion requires for warming fresh air brought in from outdoors. The arrangement of this system is shown in Fig. 2. When the window sweeping and exhaust sets are not in use, it is possible to recirculate air used for cooling the auxiliary generators, thus obtaining enough waste heat to warm the auxiliary turbine room and the fan room. In mild weather it is necessary to pump a certain amount of fresh air into the auxiliary turbine room for cooling purposes, using the blower sets discharging through the wall registers for that purpose. This air is tempered by the heaters shown in the insert in the upper left hand corner of Fig. 2. If occasion requires, this system can be used to sweep out moisture leaking from the shaft packing of the auxiliary turbines, or otherwise escaping into the room. The capacity of these sets is sufficient to furnish five air changes per hour in the auxiliary turbine room.

Steam Requirements

A summary of the surface and steam requirements of the direct radiation and ventilating sets is given in Table 1. The figures given are for the ultimate plant, i. e., for a plant containing six 50,000 kw. main units and six 4,000 kw. auxiliary turbo-generators.

TABLE 1—SUMMARY OF RADIATION AND STEAM REQUIREMENTS FOR BUILDING HEATING AND VENTILATING TURBINE HOUSE AND BOILER HOUSE, DELRAY P. H. 3—ULTIMATE SIZE

	Square Feet Direct Radiation	Pounds of Steam Per Hour
Heating System:		
Turbine House	15,500	3,875
Boiler House	14,500	3,625
Total	30,000	7,500
Ventilating System:		
Window Sweeping		5,000
Auxiliary Bay		11,000
Total		16,500
Heating Plus Ventilating:		
Heating	30,000	7,500
Ventilating, Equivalent Direct	66,000	16,500
Total	96,000	24,000

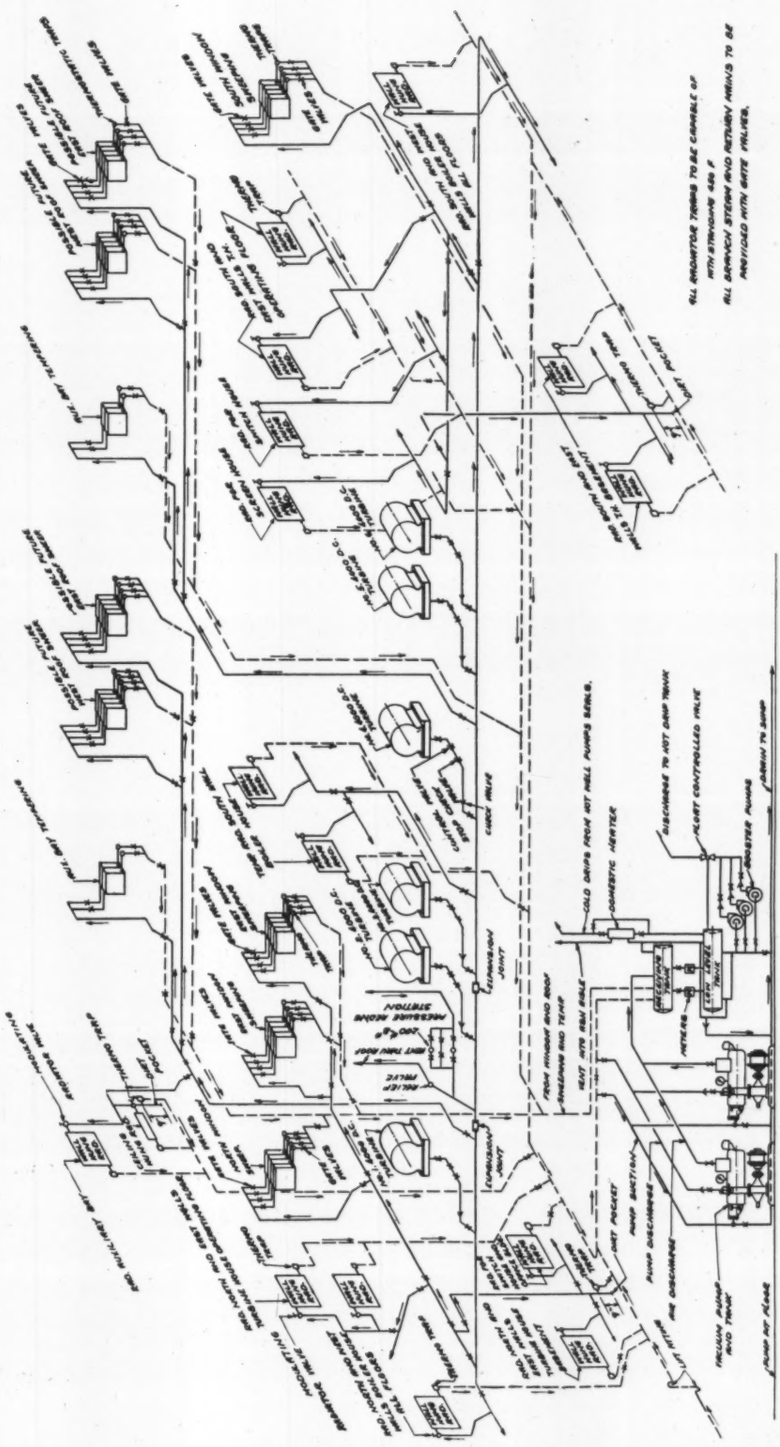


FIG. 4—LAYOUT OF THE HEATING SYSTEM. RETURNS ARE METERED BEFORE MIXING WITH OTHER PLANT DRIPS

General Features of Heating System

A line diagram of the building heating system is shown in Fig. 4. Steam for building heating is extracted from the house service turbines under normal conditions, with a reserve connection through reducing valves from the plant saturated steam header when bled steam is not available. The normal pressure in the heating header is eight pounds gage, which is maintained constant by pressure regulators at the bleeder outlets of the auxiliary turbines. Pressure in the turbine stage from which steam is bled varies from eight pounds gage at quarter load up to seventy-five pounds gage at rated turbine load. At very light auxiliary turbine loads there is insufficient stage pressure to furnish bled steam for building heating and it is then necessary to take live steam through reducing valves from the plant saturated steam header. This condition will exist only during the early life of the plant, and as load on the auxiliary turbines increases there will be ample stage pressure for heating purposes.

Considerable superheat is present in the bled steam coming from the auxiliary turbines, enough in fact to give a total temperature slightly above 450 F at rated load on the low pressure side of the pressure regulator at the bleeder outlet on the turbine. The amount of superheat present varies with turbine load, the total temperature mentioned above corresponding to rated load, while at lesser loads the temperature is somewhat decreased. While some of this superheat is dissipated by heat loss through the pipe covering of mains and branches, enough remains to preclude the use of any heating appliances put together with ordinary solder.

This has to be taken into account in selecting drip and return traps, radiator valves, extended surface copper radiation and the like. Satisfactory commercial articles for these purposes have been obtained and no trouble is experienced on this score at the present time, although some grief was encountered in earlier plants before suitable articles were developed. Drip and return traps are of the corrugated bellows type using water vapor as the expansive medium, and put together with high temperature solder which can withstand 450 F. Where unit heaters are used they are of the cast aluminum or steel coil types. Cast iron radiation has given no trouble.

No inconvenience is experienced with people being burned by getting in contact with radiators. Steam in the radiators apparently behaves much the same as saturated steam at the same pressure, and the radiators do not seem appreciably warmer to the touch than if they contained saturated steam at 5 to 10 lb. gage. All calculations for radiator surface are made on the basis of saturated steam at 5 lb. gage; which has been found to work out satisfactorily in practice. One, two and three column radiators and wall radiators are used. Vacuum pumps are installed in duplicate.

Correcting Faulty Design or Construction of Refrigeration Piping

By C. T. Baker

Waste, to be prevented, must first be realized. Very often, a few changes in a refrigerating piping system, a little more thought and care given to its construction and design, will result in a saving in operating expenses through decreased pumping costs, increased efficiency, or longer life. The author points out many of the causes of waste and the methods of correction

CAREFUL surveys and investigations made by the writer during the past several years have shown that in all too many instances the loss in capacity and waste of energy due to faulty design and construction of piping systems in numbers of refrigerating plants were very great and in several cases the energy losses were appalling.

Being particularly anxious to ascertain the reasons for such conditions, I endeavored to inquire into the several situations sufficiently, if possible, to find the answer to these questions.

Management Unaware of Piping Conditions

The first fact which came to light in most cases was that the management was not aware that the condition of its piping systems was other than correct and therefore did not realize that it was being penalized by such piping monstrosities as were found to exist.

The next fact developed was that the pipe work, for most part, had not been considered of sufficient importance to warrant careful study and design, but, on the other hand, the whole matter had been left to persons more or less ignorant and incompetent, who had only a dim knowledge of cause and effect gained from a too-limited experience and who knew a lot of things about the economics of piping systems that "were not so."

It was found also that, in many instances, the weight of the fluid or gas necessary to be circulated for a given condition had never been calculated; consequently the pipe sizes selected to meet the requirements were based on rule-of-thumb practices entirely.

In this article, effort is made to indicate some of the pitfalls necessary to avoid in piping systems if maximum economic results are to be obtained.

Selection of Pipe and Fittings

Too often, the proper kinds or classes of pipe and fittings are not selected for the conditions which are to be met. Neglect of this important matter is sure to prove expensive in the long run and often results in serious interruption of service.

Improper selection of pipe for a given purpose may, for instance, be out of all proportion in first cost to the service requirements. For example, it is still the practice of some engineers to specify extra heavy pipe for ice freezing tank coils, brine tank cooling coils, and similar service. This is not only expensive in first cost, but expensive in operation as well, since the greater wall thickness of extra heavy pipe, as compared to that of selected standard full weight pipe, retards the flow of heat between the ammonia and brine, resulting in a slower rate of evaporation of the refrigerant and a consequent slower rate of brine cooling.

Since the amount of heat that may be absorbed and carried away is a function, among other things, of the amount of ammonia evaporated per unit of time, it follows naturally then that any unnecessary wall thickness interposed between the cooling medium, ammonia, and the medium to be cooled, brine, is sure to react unfavorably on capacity.

Ammonia evaporating pressures in ice freezing tanks, etc., will range, depending upon conditions, from 15 to 25 pounds per square inch gage, while for brine cooling service, evaporating pressures seldom reach 40 pounds pressure.

Since the safe working pressure of 1¼-in. standard full weight pipe is greatly in excess of that required for the class of service described, it is evident that the engineer who insists on extra heavy pipe for this service is paying a heavy penalty in dollars and cents.

As to whether or not extra heavy pipe should be used, this will depend upon the pressures carried and the pipe diameters called for. That there are many proper applications for this class of pipe in refrigerating and ice plant service is of course admitted, and when and where the conditions warrant, such pipe should be specified.

For example, where the character of the condensing water is such as to cause abnormal deterioration of piping, extra heavy pipe may be used to an advantage in the construction of atmospheric ammonia condensers, where this type of condenser is used.

Another proper application of extra heavy pipe is in the construction of pipe headers for ammonia condensers of various types, including both the gas and liquid headers. Oftentimes, too, it is advisable to use extra heavy pipe in the construction of the gas discharge line between the compressor and condensers.

It frequently happens also that very close center return bends are needed in fabricating evaporator coils for freezing tanks. Where the pipe must be bent as close as 4½-in. centers, it has been found advantageous to use extra heavy pipe to form the bends. The sections of pipe used to form such bends are welded to the runs of standard full weight pipe making up the stands of coils.

On 1¼-in. pipe bent closer than 6-in. centers, extra heavy pipe sections should be used rather than standard

full weight, since such pipe often will not prove satisfactory.

Choice of Return Bends

Still another example fairly typical of improper selection in the matter of pipe fittings is to be found in the choice of return bends employed in the construction of cooling coils where brine is used as the cooling medium.

In order to save in first cost, the owner or engineer will often use return bends of very close centers, rather than bends of larger centers, with the result that the pipe sections which make up the coils are nested so closely together that frost soon accumulates in sufficient quantities to bridge across the space between adjacent pipes, thus forming a solid mass of frost across the entire width of the coil.

FIG. 1—A CEILING COIL WITH PIPES ON TEN-INCH CENTERS. NOTE THAT THE AIR MAY CIRCULATE FREELY

Wherever this condition exists, it is very difficult to maintain proper room temperatures, since the frost completely prevents the circulation of air between the individual pipe of the coils, and also, the frost acts as an insulator, retarding the flow of heat from the brine to the air. Fig. 1 illustrates a ceiling coil with pipes on 10-in. centers. Note that air may circulate freely.

In addition to the disadvantages resulting from the use of the close pattern return bend as mentioned, their use also adds somewhat to the friction losses in the coils. This, of course, is reflected in an increase in the pumping cost.

Generally, and unless there are certain very definite structural limitations, it will pay to use return bends of from 10 to 12-in. centers in cooling coil construction, regardless of whether the coils are used for brine circulation or for direct expansion ammonia. Of course, there are notable exceptions to this rule, such as special type bunker construction and small ice-cream hardening rooms where facilities are provided for quick defrosting.

Recently it was decided to replace a steam driven brine pump with an electric driven centrifugal pump for circulating brine in a number of small meat curing rooms.

As it was necessary to determine the head under which the existing pump was operating, a pressure gage was placed on the pump discharge. Very much to the surprise of everyone, the gage indicated a pressure of 90 pounds with the steam pump operating at normal speed. An investigation of the piping arrangement disclosed that several hundred feet of 1¼-in. pipe which constituted the coils located on three walls of some four rooms had all been connected in series and also that return bends of 4-in. centers had been used.

As the cost of pumping brine with purchased current at the high head observed would have been out of proportion to the revenue received from the refrigeration performed, it was evident that a change in the coil arrangement was necessary.

After the coils in each individual room had been equipped with headers permitting parallel flow of brine rather than series flow, and with the return of the coils of each room terminating in a larger discharge line, it was found that the discharge pressure was reduced from 90 pounds to 20 pounds per square inch, or a reduction of 70 pounds.

In the case referred to, the pump replaced by the electric driven unit had been operating from 3 to 5 months per year for a period of nearly ten years previous to the electric drive.

It is evident that in this period of time many thousands

of pounds of steam were unnecessarily consumed at the expense of the coal pile.

The Necessity for Care in Making Joints

At the present time, most of the refrigerating plant piping, embracing the ammonia, and brine systems, water supply, etc., are made up with some one of the several available forms of joints. In many instances the pipes are welded.

Care should be taken in the welding of pipe to avoid trouble due to ignorance and carelessness. This, of course, also applies to the use of fittings.

In a great many cases, both the high and low pressure ammonia piping employ what is known as the sweat-soldered joint for making up coils and for connecting sections of pipe.

The sweat joint is made by tinning the threads of both the pipe and fittings, then screwing the fitting on to the pipe while both the end of the pipe and the fitting are at about the melting temperature of the solder.

After the fitting has been screwed on to the pipe as far as it will go, the fitting recess is then filled flush with solder. This makes an excellent, though sometimes rather expensive, type of joint which insures against thread leakage and the solder which fills the recess of the fitting protects the pipe threads that otherwise would be exposed.

Some time ago, the writer had occasion to examine a lot of pipe coils that had been in service for some years, but which were removed to permit certain building changes to be made.

An examination of the removed coils and connections showed that the joints had been made up by soldering as described above. An inspection of a 20-foot section of pipe which had connected one of the coils to the main suction header disclosed that there had been left, in one end of the pipe, a ring of solder which apparently had been formed by the placing of the section of pipe in the solder pot where the temperature of the solder was not correct at the moment and which allowed a collar to be formed near the pipe end. Since this collar had never been noticed, the section of pipe in question was placed in service where it had been for some years and where the effective area of the coil outlet at this point had been reduced at least 50 per cent—an unusual condition to be sure but still one of those things that can and does happen when care is not exercised in making solder joints.

The flange and the pipe end had both been properly tinned. Likewise, the flange had been screwed up properly on the pipe. The thing, however, which caused the trouble was carelessness in finishing the job.

In still another case where it was impossible to make a direct expansion coil frost, it was found, upon disconnecting the coil from the headers and taking the sections apart, that one of the pipe sections was completely stopped up with a ball of solder which had adhered to the interior surface of the pipe when it had been immersed in a pot of solder of too low a temperature. The cost of dismantling the coil in question and locating the trouble was considerable.

These two cases immediately suggest the necessity for a careful examination of each section of pipe before erection, since the time required to examine each individual section is negligible. In this connection also, it will always pay to blow out thoroughly all coils and connections with compressed air, wherever possible, before placing them in service.

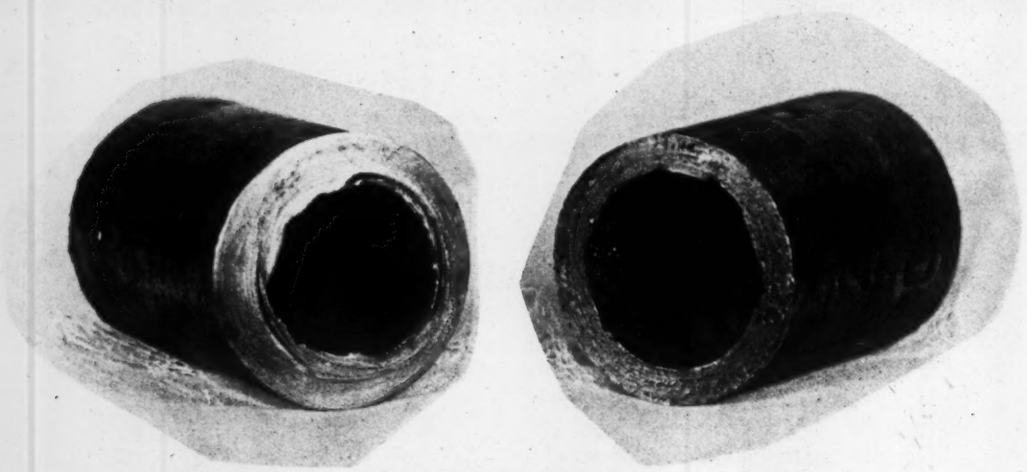
Cutting Pipe

One of the harmful practices of some pipe erectors is that of using improper pipe tools when fabricating pipe work. For example, some pipe fitters seldom use a pipe reamer for removing burrs formed by pipe cutting tools. This is particularly unfortunate in the case of small diameter pipe, since it often happens that pipe ends are greatly reduced in area by such burrs or collars formed by the use of some cutters.

It is not difficult to visualize the loss in carrying capacity of, say, a half-inch or three-fourths-inch pipe line of several hundred feet where the ends of each pipe section are reduced in area by a heavy internal ring or collar.

Fig. 2 illustrates very clearly just what happens when pipe ends are not reamed. This illustration shows two short sections of three-fourths-inch pipe, one of which

FIG. 2—THIS ILLUSTRATION SHOWS CLEARLY WHAT HAPPENS WHEN PIPE ENDS ARE NOT REAMED. ONE OF THESE TWO SECTIONS OF THREE-FOURTHS INCH PIPE WAS CUT BY A HACK-SAW; THE OTHER BY A THREE-WHEEL PIPE CUTTER



was cut with a three-wheel pipe cutter and the other with a hack saw.

Too much emphasis cannot be laid on the necessity for care in removing all such obstructions in pipe lines before erection. This is important in any case, whether the pipe be of large or small diameter. Tapered pipe reamers should be a part of every pipe fitter's equipment and he should be made to use this tool.

The Use of Litharge and Glycerine as a Joint Cement

In a great deal of refrigerating plant work, ammonia joints, in place of being sweat-soldered as previously described, are made up with a mixture of glycerine and litharge which forms a pasty mass and which is used on the threads of pipe and fittings just as other kinds of thread cement are used.

After the fitting has been screwed to the pipe, the recess is filled flush with the litharge mixture, which will harden and form a protection for any part of the pipe threads that remain outside of the fitting proper.

The glycerine and litharge mixture is a very excellent thing to use in making up pipe joints, but like many other good things it is oftentimes improperly used, and where this occurs, trouble will follow.

In screwing fittings on small diameter pipe, the pipe fitter will frequently use the litharge mixture on both the threads and the pipe in very liberal quantities.

After the fitting has been screwed home, it will be observed that a ring or collar of litharge has formed around the end of the pipe. If this is not removed it will quickly harden and later, when the pipe is placed in service, its diameter will have been so greatly reduced as to make it almost impossible for the pipe line to carry sufficient refrigerant necessary to meet operating conditions.

Some years ago, a large hotel in New England experienced a great deal of difficulty in refrigerating a number of small brine-cooled refrigerators located on all typical floors of the building.

After considerable annoyance and expense, and some spoilage of goods, the ¾-in. connections between the brine mains and the 1¼-in. coils in the refrigerator were taken down and in every case were found to be almost entirely closed up with a hard ring of litharge.

After removing the obstruction and replacing the connections, no further trouble was experienced. It is important, therefore, when using litharge, particularly with the small diameter pipe, that any surplus material in the form of a ring or collar inside the end of the pipe be removed before the pipe is erected.

Some engineers require that each section of pipe be swabbed out just before being placed in position in order to remove sand, grit and other foreign matter.

This will require a little more time, but it is something that will pay dividends, since it will not only insure the removal of foreign matter but, in the case of ammonia piping, will prevent foreign matter reaching the compressors and causing trouble due to scoring.

Flange Bolts

The writer has found on several occasions that mechanics, in erecting both ammonia and steam piping where flanged fittings have been used, have used flange bolts that were not only of much smaller diameter than called for, but also in some instances the bolts were too short, allowing only from 30 to 50 per cent of the nut to be screwed on to the bolt after it was in position.

Bolts used on piping systems, particularly where pressures carried are high, as in the case of ammonia and carbon dioxide refrigerating plants, should be of the correct diameter. Bolts that are just small enough to pass through the bolt holes in the flanges should be used. If bolts are too small in diameter, they are very likely to pull in two, causing an accident and the serious loss of ammonia or other refrigerant.

About a year ago, an accident occurred in an electric-driven ice plant of about 50 tons capacity in which the main discharge line opened up at a pair of flanges. This accident resulted in a loss of considerable ammonia and a shut-down of several hours. An investigation disclosed that the bolts used for this particular pair of flanges were much smaller in diameter than should have been used and, also, the material from which the bolts were made was of uncertain quality. Being in need of a set of bolts quickly, at the time the original pipe line was installed, a set unsuited for the service was obtained. These bolts being at fault, both as regards size and quality of material, could not stand the strains of service and failed without warning.

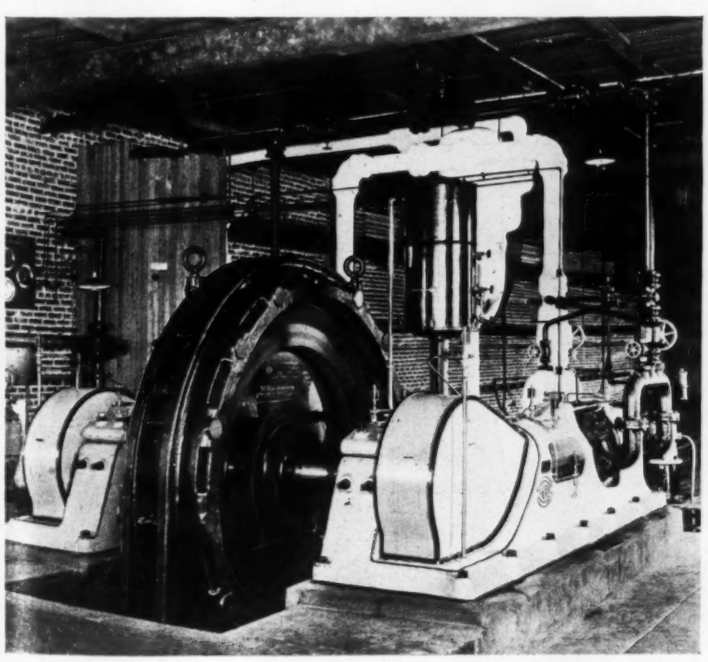


FIG. 3—A FIFTY-TON AMMONIA COMPRESSOR DRIVEN BY A SYNCHRONOUS MOTOR IN AN ICE PLANT

Pipe Arrangement Important

As mentioned previously, a great deal of piping in ice and refrigerating plants is installed without a knowledge of the requirements which should govern. Then again, much of the pipe work, particularly water lines and cold brine lines, is constructed without reference to proper arrangement,

FIG. 4—VIEW OF THE TANK ROOM FROM DUMP PLATFORM IN AN ICE PLANT

consequently it happens frequently that pipe lines contain an unnecessary number of right angle elbows and are not as short and direct as conditions would permit. Too often, small pipe lines are supported with hay wire hangers which permit the lines to sag and settle and impose unnecessary strains on both the pipe and fittings.

Allowable Gas Velocities

For some years considerable discussion has occurred at times regarding the optimum or economic gas velocities for suction and discharge pipes leading to and from ammonia compressors.

There still exists differences of opinion on this subject, but all authorities are agreed that gas velocities should not be excessive and that therefore pipe diameters should be sufficiently large to hold pressure losses within the limits of good commercial practice.

A gas velocity in ammonia discharge lines of 6,000 to 6,500 (or even 7,000) feet per minute is considered by many as satisfactory, while for suction lines 4,000 to 5,000 feet per minute is generally accepted as representing good practice.

It is particularly important that suction pipe areas be ample; otherwise a pressure drop between the evaporators and the compressors is inevitable.

Such pressure drops, even if only two or three pounds, will materially affect both the capacity and the horsepower consumption per ton output.

Ample proof of this statement is found in the tabulated results of carefully conducted tests which have been made to determine the b. hp. per ton of refrigeration at different suction pressures and at constant condensing pressures.

The writer has found many cases where ammonia suction lines have been so poorly laid out as to cause excessive pressure drop between evaporator outlet and compressor inlet, and in each instance it has developed that such lines had been constructed without any thought or consideration of the actual gas carrying requirements which obviously should include such items as weights, velocities, pressure drop, etc.

It is a noteworthy fact, however, that present day pipe line practice is a great improvement over that of a few years ago. This spells progress and economic gain.

Cotton by weight is less hygroscopic than either wool, silk or flax but it is more sensitive to atmospheric variations. Cotton fibers under the microscope look like flattened, twisted tubes. In spinning, a natural twist must be given to the fibers so that they will seize each other and grip tightly together. But the inner fiber is surrounded by a coating of viscous waxy matter and the whole is enclosed in an outer sheath. The manufacturing process is affected by the condition of this waxy material. When it is cold, it is hard, and as the temperature increases it becomes softer and melts at about 180 F. It is obvious that there is one temperature at which the fibers will knit most closely and strongly together. Moisture also affects their flexibility. A dry splinter of wood is easily broken, but when it is full of moisture, it can be bent in any direction without breaking. Similarly, cotton fibers that are too dry are easily broken and when too wet, they can be pulled apart easily.

It is also important that the vapor in the air be finely divided. It is possible to put moisture in the air in such form that the particles are so large that they will drop down like the drops in a fine rain. Static is avoided by means of moisture in the air. What is termed "fly" or floating particles of cotton in the air is lessened by the introduction of moisture. Sometimes the amount of "fly" that is swept up from the floors of a textile mill constitutes considerable loss.

The Value of Air Conditioning *for* Persons Doing Physical Work

in Dollars *and* Cents

By Malcolm Tomlinson

FOR over six years the research laboratory of the *American Society of Heating and Ventilating Engineers* has been investigating the effect on human comfort of temperature, humidity and air motion. A wide range of comfort information has been published. The caliber of this work is such that those who have contributed to the research efforts deserve the warmest praise. This work progresses because the demand for more and more information is pressing, also because the air conditioning field is widening.

Information Needed on Air Comfort and Its Effect on Amount of Work

Up to the present time, there has been a lack of real information on the value of air comfort in terms of a day's work. No doubt more research is needed before such values can be placed upon a scientific basis. At the same time, until something tangible is published, the use of comfort data will be restricted. This will not only retard progress in the air conditioning industry, as well as the fields which benefit directly from such service, but is unfortunate so far as the worker is concerned.

To meet the situation described, the available comfort data have been surveyed and those which are pertinent have been analyzed. The results proved that it was possible to approximate the value of human comfort for production purposes within certain limits. Since the need for even approximate data is great, the sources of information, the various steps in the analysis and the proposed method for evaluating human comfort are herewith presented.

In 1925 and 1926, the *A. S. H. V. E.* research laboratory issued reports on the results of human comfort tests in atmospheres of high and low temperatures, respectively, and various humidities in still and moving air. The subjects were lightly clad and in normal health. This latter factor should be carefully noted, since it is evident, from medical research experience, that persons with diseases of the nose, throat and lungs require a degree of comfort different from those who are in health. A discussion of these factors will be found in "Air Conditioning for Diseases" in the February, 1929, HEATING, PIPING AND AIR CONDITIONING.

Zones of Comfort that Vary with the Seasons of the Year

Throughout the present article, the term "effective temperature" will be used. It may be defined as the true measure or index of a person's feeling of warmth or coldness under all combinations of temperature, relative humidity and air motion. It applies, as charted from test observations, only to average human beings in normal health. For illustration, an effective temperature of 66 F, under any condition of temperature, humidity and air motion within its range, gives best human comfort for winter. Due to the changes made in clothing, the greatest comfort for the summer season is found with an effective temperature of 70½ F. It, therefore, follows that there are zones of human comfort which vary with the seasons of the year. In order that the relation between various effective temperatures may be seen, several examples for persons normally clothed and healthy will be given:

Dry Bulb	Still Air		700 F.P.M.	
F	100% R.H.	20% R.H.	100% R.H.	20% R.H.
66	66	61.2	53.8	52.4
80	80	70.7	71.7	65.6

For further information the reader is referred to the very complete table of effective temperatures to be found on page 96 of the 1925 *Transactions of the A. S. H. V. E.*

Tests Conducted by A. S. H. V. E.

The first set of tests, in which the subjects worked in atmospheres of high temperatures until exhausted, is of great importance. The 1926 work tests involved temperatures which were lower and the subjects worked only for two and one-half hours. These latter tests could have been conducted until the subjects were exhausted, but the objects sought did not include prolonged effort. The work in both tests consisted of raising and lowering a fixed weight through a given height with reasonable rest periods. The tabulated results may be seen in the *A. S. H. V. E. Transactions* for the years noted and will not be repeated here. The rate of work and the effective temperatures prevailing have been calculated and will be found in Tables 1 and 2, where part of the test data are noted. The total work performed in the first set of tests is plotted in Fig. 1 against the dry bulb temperature. This chart is reproduced from the published report. No charts will be shown for the second set of tests for the reasons noted above. Nevertheless, the interested reader will find it worth while to study the data from the second set of tests in connection with the information developed in this article.

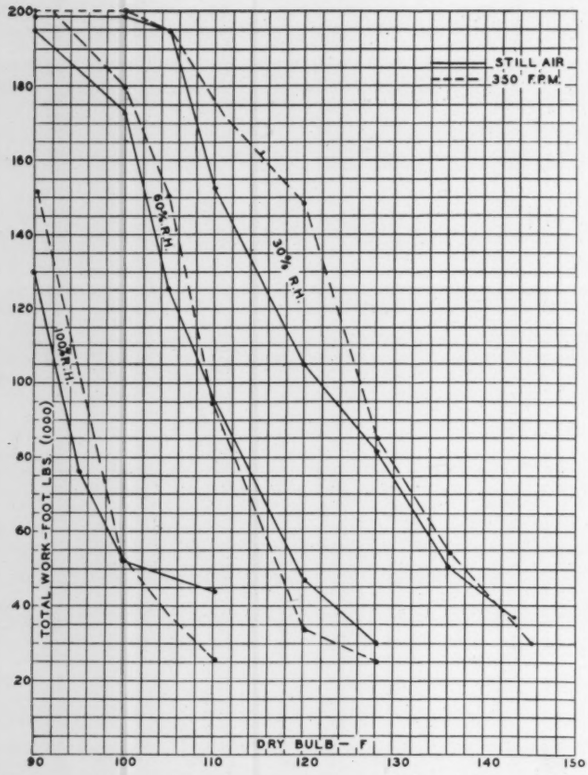


FIG. 1—WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES. THE SUBJECTS RAISED AND LOWERED A FIXED WEIGHT THROUGH A GIVEN HEIGHT UNTIL EXHAUSTED

This Approximate Method Applies to Physical, Not Mental Effort

The people who took part in these work tests had no occasion to exercise mental effort. How much effect comfortable or uncomfortable atmospheric conditions have on the human mind has never been adequately studied. We do know that certain atmospheric conditions, where the air is foul or above normal in temperature, produce drowsiness. It is also known that air which is not in circulation, though "fresh" and normal in temperature, has a slowing up effect upon the activity of the mind. Therefore, in foul air, the value of human comfort will be greater than will be apparent from the data presented in this article. Furthermore, it is probable that mental fatigue has a considerable effect on physical fatigue. As this latter factor did not enter into the tests which are about to be discussed, it follows that the approximate method here proposed for evaluating human comfort does not apply to work which requires mental effort.

From Fig. 1, two facts are quite evident. One is that the test data for still and moving air at each given humidity are quite close. The other is that, while 200,000 foot pounds appear to be the maximum work which can be performed at 30 per cent relative humidity under the test conditions, the maximum for 60 and 100 per cent had not been reached. Apparently, the maximum for 60 per cent would be higher than for 30 per cent.

When the same work performed is plotted against effective temperature, as in Figs. 2 to 4, the effect of humidity and air motion is included with the effect of temperature. Here it is apparent that a definite relation

TABLE 1—STILL AIR-SUBJECTS LIGHTLY CLAD

Temp.—F		Rel. Hum. Per Cent	Eff. Temp. F	Rate of Work Per Hr. Ft. Lbs.
Dry Bulb	Wet Bulb			
90	67.4	30	76.3	87,300
100	74.4	30	82.6	89,800
105	78	30	85.7	85,800
110	82.1	30	88.8	86,500
120	88.9	30	95.2	74,000
128	95	30	99.9	90,700
136	100.7	30	104.8	90,700
143	107.5	30	109.1	91,000
90	78.3	60	82	88,100

exists between maximum work (or fatigue) and human comfort.

Tables 1 and 2 record the dry bulb temperatures, relative humidities, effective temperatures and rates of work in foot pounds per hour for the first set of tests. It will be noticed at once that the rate of work, with few exceptions, was very uniform, also that the average rate for the various humidities varied slightly. It is not possible to furnish the rates of work for 60 and 100 per cent humidity in still air, as the data for these conditions were not published by the research laboratory. Nevertheless, the information obtainable clearly indicates that, although the period of work varies from 15 to 137 minutes, the rate of work is consistent.

When the total work, found in Fig. 1, is plotted against the time noted in the test reports, another definite relationship is evident. Figs. 5 to 7 give these curves for still and moving air (350 f.p.m.). The few data which are not consistent are likely to be the result of errors of observation or of uncontrolled factors which affected

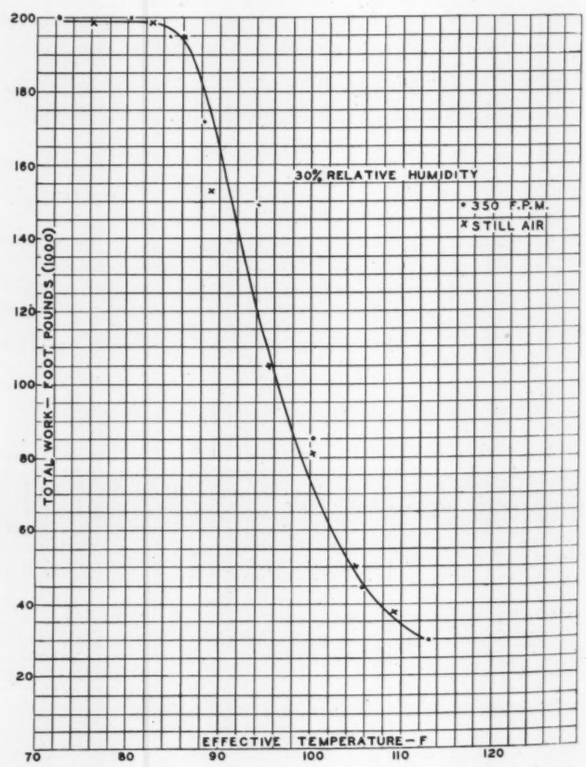


FIG. 2—WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES PLOTTED AGAINST EFFECTIVE TEMPERATURE FOR 30 PER CENT RELATIVE HUMIDITY

TABLE 2—350 F.P.M.-SUBJECTS LIGHTLY CLAD

TEMP.—F		REL. HUM. PER CENT	EFF. TEMP. DEG. F	RATE OF WORK PER HR. FT. LBS.
DRY BULB	WET BULB			
90	67.3	30	72.4	87,300
100	74.4	30	80.4	90,000
105	78	30	84.6	86,000
111	82.2	30	88	83,400
120	89	30	94.1	134,300
128	95	30	100	88,500
136	100.7	30	105.4	97,000
145.4	107.1	30	113.1	48,500
90	78.3	60	76.8	91,200
100	87	60	87	88,600
105	91.5	60	91.6	75,600
110	96	60	97.2	84,300
120	104.8	60	109	83,500
128	112	60	120	97,800
90.4	90.4	100	86.4	81,400
95	95	100	93	81,100
100	100	100	100.2	32,650
105	105	100	107.7	126,700
110	110	100	118	83,700

the results. There is, of course, nothing surprising about this relation. It is due to the fairly uniform rate of work which was noted in the two tables. Two important facts can be deduced from these three figures. One is that the rate of work was such as to exhaust a normal human being, lightly clad, in from 15 to 137 minutes, dependent on atmospheric conditions. The other is that there apparently was no difference between the results obtained under still and moving air conditions. In fact, the differences due to various humidities were slight. This is consistent with the curves shown in Figs. 2 to 4.

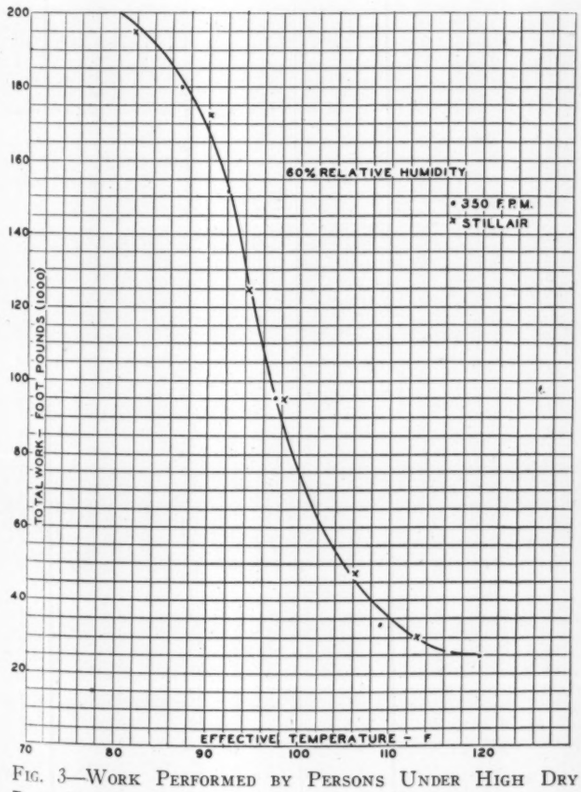


FIG. 3—WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES PLOTTED AGAINST EFFECTIVE TEMPERATURE FOR 60 PER CENT RELATIVE HUMIDITY

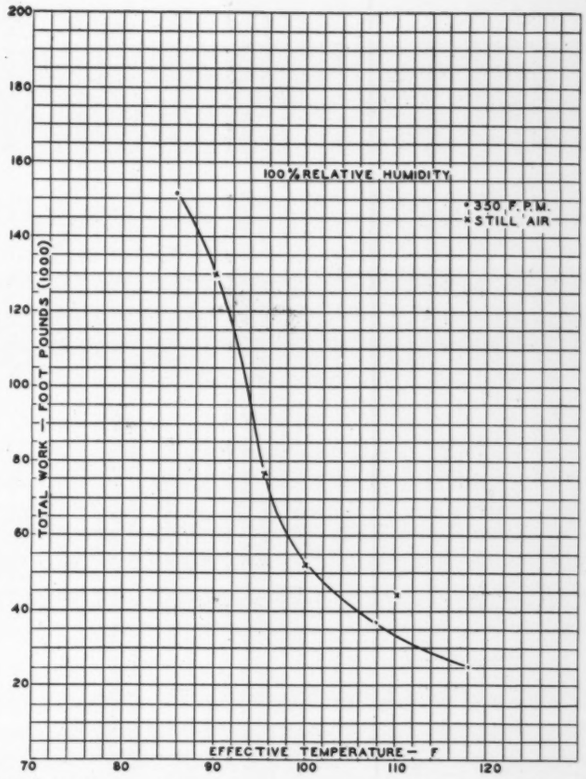


FIG. 4—WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES PLOTTED AGAINST EFFECTIVE TEMPERATURE FOR 100 PER CENT RELATIVE HUMIDITY

If the plotted data from Figs. 2 to 7, inclusive, for each particular step in total work is obtained and replotted in three separate humidity curves with time as the ordinate and effective temperature as the abscissa, a new set of relations is obtained for comfort based on time. In Fig. 8 these curves will be seen. There is evidence, in the way they cross one another, that more test data are needed to straighten out the curves. Nevertheless, with the data available, these curves give a very clear picture of comfort relations under working conditions. Here is the time limit for the average human being under a wide range of effective temperatures.

Each of the three curves in Fig. 8 can now be plotted by itself at regular intervals from the other curves. This has been accomplished in Fig. 9, where relative humidity curves for 40, 50, 70, 80 and 90 per cent have been interpolated. All of these humidity curves can now be connected by lines of total work based on test data to be found in Figs. 2 to 4. Separate effective temperature scales are furnished for each humidity curve as ordinates. With this information, a chart is now available for figuring the value of human comfort in that region of atmospheric conditions which is most uncomfortable.

A careful examination of the source data in Figs. 1 to 8, inclusive, will show that Fig. 1 follows the test data exactly and that the other figures furnish satisfactory mean curves in which most of the test data are within the limits of experimental error. Since Fig. 8 is rigidly based on Figs. 2 to 7, inclusive, it is evident that the data in Fig. 9 furnish a very good approximation of the test data and are, therefore, dependable. The real difficulty is that the test data are meager and do not supply information in the region of normal tempera-

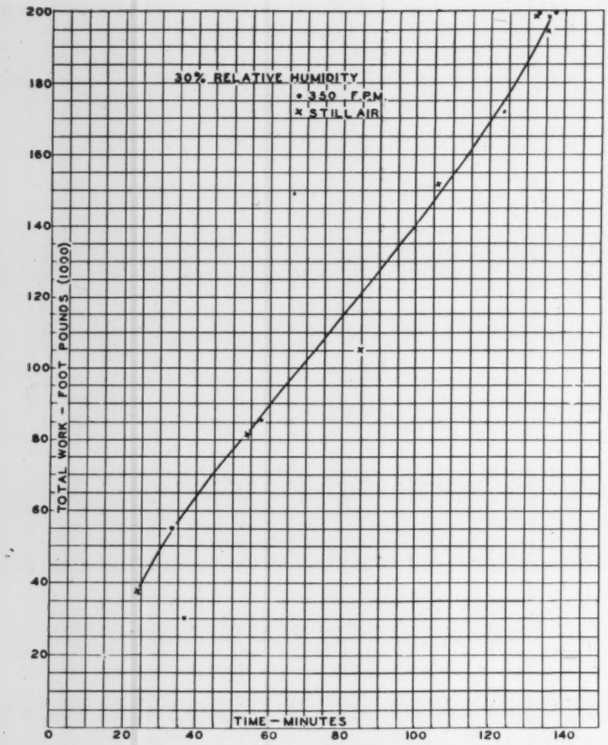


FIG. 5—TOTAL WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES AND 30 PER CENT RELATIVE HUMIDITY PLOTTED AGAINST TIME

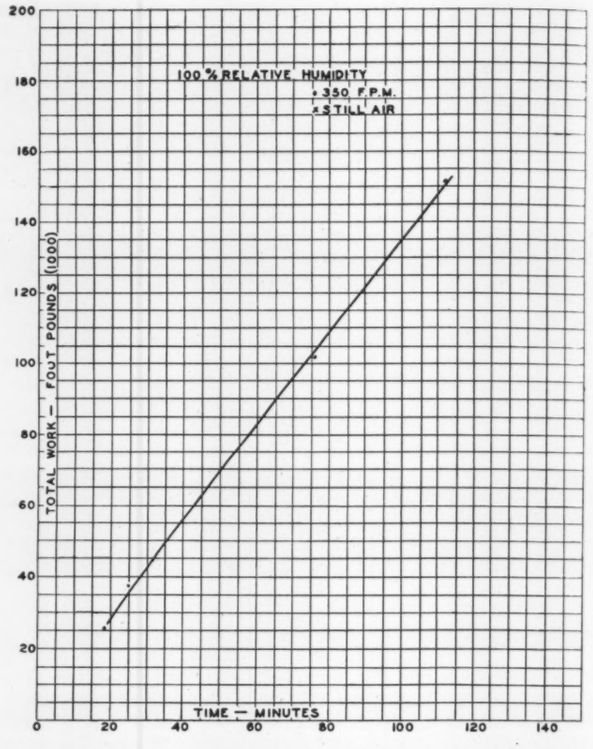


FIG. 7—TOTAL WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES AND 100 PER CENT RELATIVE HUMIDITY PLOTTED AGAINST TIME

tures. As has already been pointed out, further test information would undoubtedly result in a greater degree of consistency. For these reasons, the chart in Fig. 9 must be considered as an approximation.

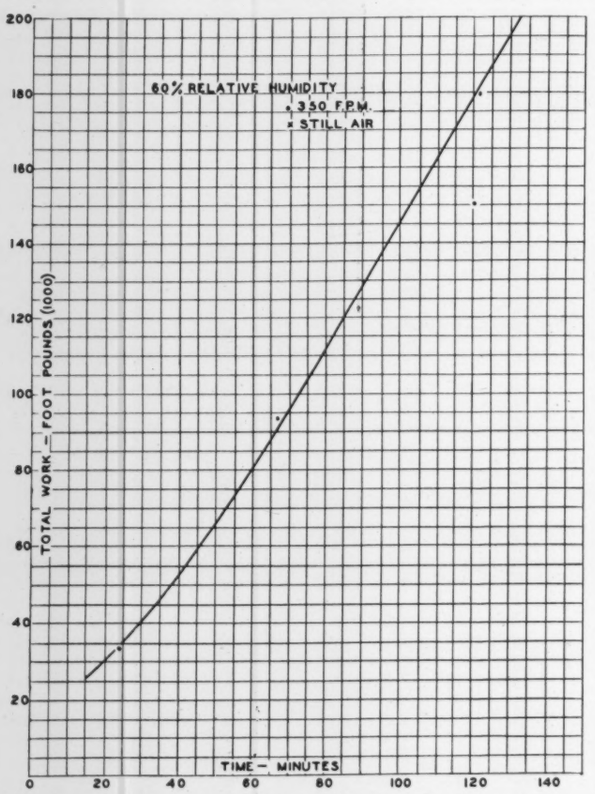


FIG. 6—TOTAL WORK PERFORMED BY PERSONS UNDER HIGH DRY BULB TEMPERATURES AND 60 PER CENT RELATIVE HUMIDITY PLOTTED AGAINST TIME

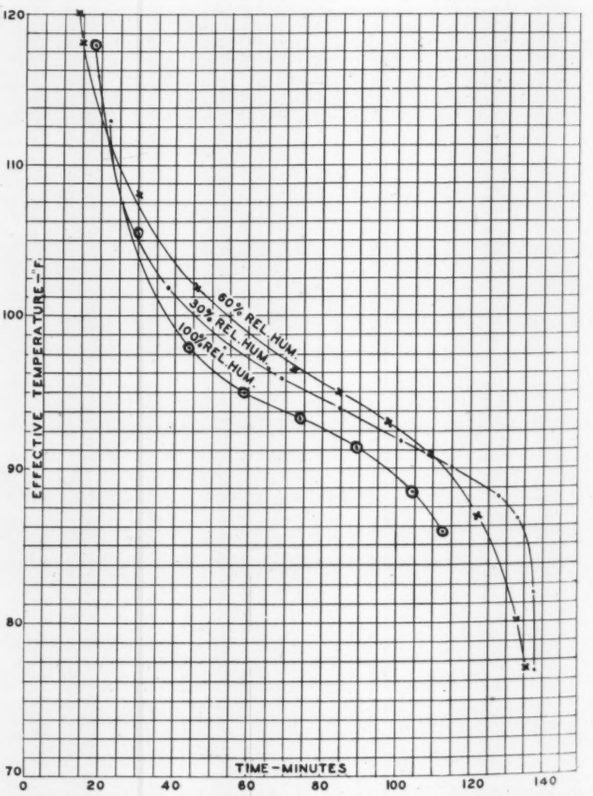


FIG. 8—TIME LIMIT FOR AVERAGE PERSONS PERFORMING PHYSICAL WORK AT VARIOUS VALUES OF RELATIVE HUMIDITY AND A WIDE RANGE OF EFFECTIVE TEMPERATURES

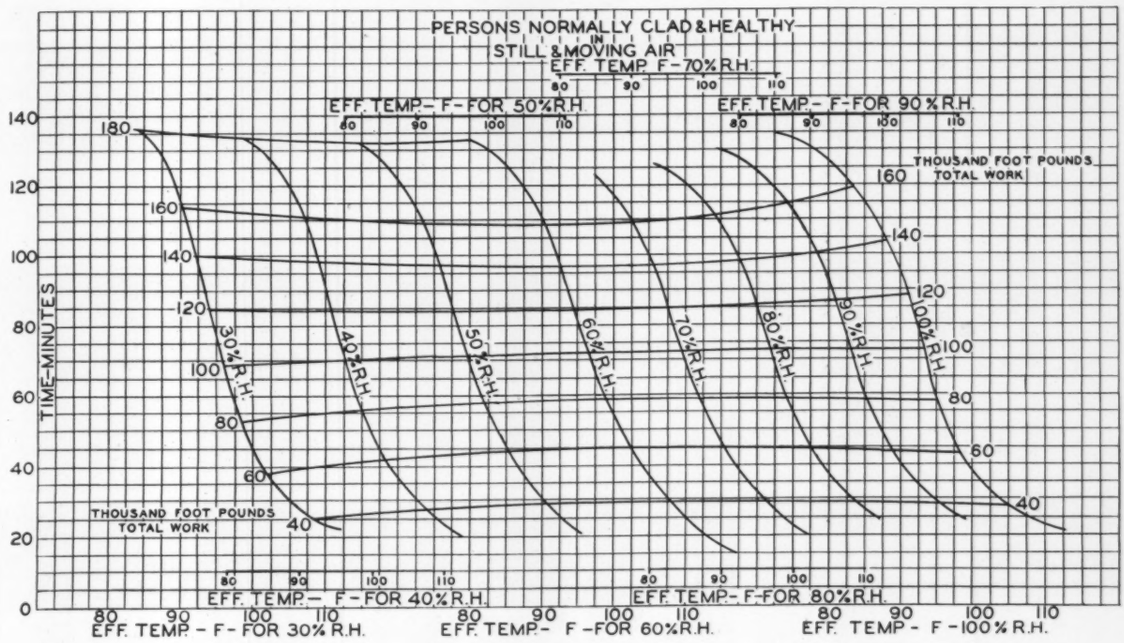


FIG. 9—APPROXIMATE VALUE OF HUMAN COMFORT IN FOOT-POUNDS

Example of Use of Information

To use this information, it is necessary to have at hand the comfort charts or tables of the *A. S. H. V. E.* so that various weather conditions may be interpreted in terms of effective temperatures. A single problem will illustrate the use of the chart. Assume a dry bulb temperature of 103 F, a relative humidity of 100 per cent and still air. From the table on page 96 of the 1925 *A. S. H. V. E. Transactions,* the effective temperature is found to be 103 F. Looking at Fig. 9, it is found that 44,000 foot pounds, under such conditions of discomfort, will result in exhaustion in 31½ minutes. If the effective temperature is lowered to 80 F, at a relative humidity of 60 per cent, 180,000 foot pounds of work could be performed in 133 minutes before exhaustion. It is now easy to calculate that, at 60 cents per hour for labor costs, it would be possible, if it were not for exhaustion, to do 180,000 foot pounds of work under the first condition for $1.29[1] and under the second condition for $1.33. If it takes 15 minutes to recover from exhaustion in the first case, it will take at least a full hour of rest to complete 180,000 foot pounds of work.[2] Therefore, this amount of work will cost, actually, $1.89, and the saving due to the increased comfort of the second condition is $0.56 per 180,000 foot pounds of work. (The term "foot pound" has been used through this article as a measure of work; it is defined, of course, as follows: When a mass which weighs one pound is moved through a space of one foot a foot pound of work has been completed. This unit is quite small, as can be seen from the fact that it takes 33,000 foot pounds of work to equal one horsepower. A man who weighs 200 pounds performs 1,056,000 foot pounds of work when he walks 5,280 feet or one mile. When a lever which weighs one pound is raised and lowered 20,000 times the work done is 40,000 foot pounds. This explanation does not take into account friction, which, in many work problems, is an important factor, but it serves to make the term clear. Many jobs can thus be quite easily expressed in foot pounds, or in horse power, as far as the work performed is concerned. It is also easy to express this work in some unit of time such as one hour. The term thus becomes foot pounds per hour.)

It is evident, from the above example, that not only can savings in labor costs be figured by the method described, but also that a way to effect such savings is by reducing the effective temperature to such a point that air conditioning will pay for itself and also that employes will have comfortable conditions. It is interesting to note that, with air moving at 200 f.p.m., the second condition of the problem would mean a dry bulb temperature of 89½ F.[3] This enables one to visualize the differences which can be obtained in working conditions. Through the new comfort valuation chart herewith presented, the factory executive can easily figure savings by increased comfort through increased output and decreased rest periods.

[1] $\frac{180,000}{44,000} \times 31\frac{1}{2} = 129$ minutes
60c per hr. = 1c per min.
$129 \times 0.01 = \$1.29$

[2] $\frac{180,000}{44,000} = 4.08$
4.08×15 minutes = approximately 1 hour.

[3] See chart facing page 96 in the 1925 Transactions of the A. S. H. V. E.

Cooling by Evaporation

The extent to which the air of a building can be cooled by evaporation depends upon the difference between the wet bulb and the dry bulb thermometer reading. If the air is very dry, the air can be cooled a great deal. If it is very moist it can be cooled very little. If the air is saturated, the air can not be cooled at all by evaporation of water.

Design of Warm Air Fan Blast Systems for Large Structures

IN general, the design of a direct-fired, hot blast system of heating and ventilating follows the same methods employed in the design of a hot blast system of steam, except direct fired units (furnaces) are substituted for the indirect radiation, or pipe coils.

Many books have been written on the subject of design, and it will be considered unnecessary to take up any of the reader's time with any lengthy discussion or treatise. The average heating and ventilating engineer is familiar with allowable velocities, etc., but the practical application of formulas and rules for ready and instant use, thereby eliminating brain splitting problems in calculus for the design of a simple duct system, is desirable.

Unit and Trunk-line Systems

One of the favorite subjects for discussion among designers of direct-fired systems of fan blast warm air heating and ventilating is the "unit system" versus the trunk line or central system. The choice in factory or industrial work depends largely upon the conditions. Where factory employes sit at benches or machines and are not bodily active, distributing ducts are often considered necessary to minimize drafts. In foundries, machine shops, auto-body works, and in industrial buildings where men are engaged in heavy or semi-heavy labor, one main duct, or if the building is not too long, one inlet from the end or center, or one central unit will often be said to suffice.

A "Ductless" System

The accompanying plans show the three types of systems under consideration. The first is the "ductless," with the direct-fired units installed in a separate building. An almost identical system to Fig. 1 is installed in an auto-body works in St. Louis, Mo. In this plant there is no trouble in maintaining 70 degrees in the two far corners of the building in the coldest winter weather. However, in the plant, the movement of the return air

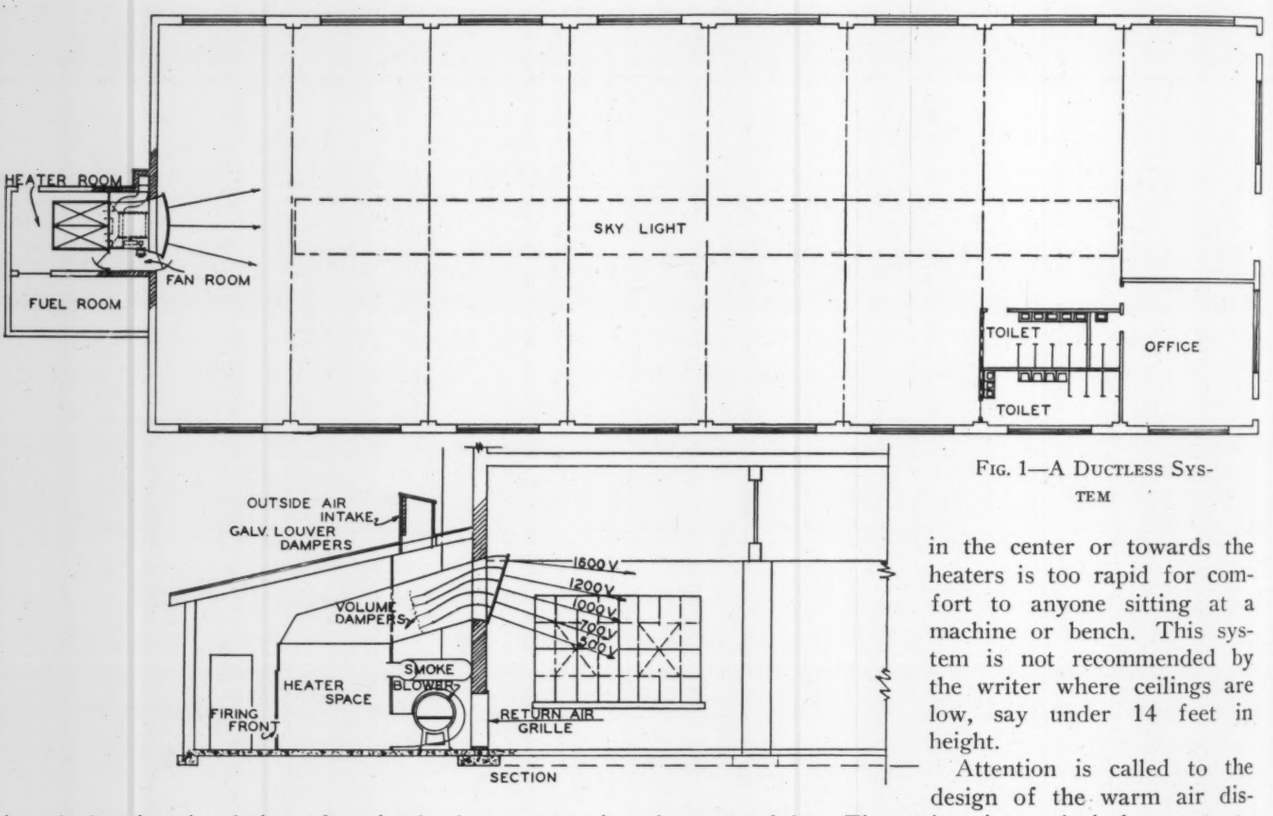


FIG. 1—A DUCTLESS SYSTEM

in the center or towards the heaters is too rapid for comfort to anyone sitting at a machine or bench. This system is not recommended by the writer where ceilings are low, say under 14 feet in height.

Attention is called to the design of the warm air discharge or inlet. The writer is particularly averse to "trained arrows," but those shown give a general idea of the results obtained by this design. No hard or fast rule for definite velocities is known, but, in the writer's opinion, they may be based on 100 ft. of velocity for every 10 ft. of distance the air must be forced. It is obvious that some of the air does short circuit, and must be compensated for in the c.f.m. supplied. The amount added to the requirements for this short circuiting in the above system was 10 per cent more than required for the duct system.

A Unit System

The unit type of system is shown in Fig. 2. Such a system must be oil or gas fired if the wheeling in of

By Platte Overton

fuel or the removal of ashes is objectionable, and may be automatically controlled. When the room temperature drops lower than that desired, the oil burner or gas burner starts. When the plenum chamber reaches a temperature of 120 to 150 degrees, a thermostat starts the fan. When the desired temperature in the space heated is reached, the oil burner or gas burner cuts off and the fan continues to run until the plenum chamber drops to 80 or 90 degrees. Here again, if short circuiting be experienced due to the proximity of the discharge and the fan inlet, it can be compensated for as outlined above.

No air conditioning apparatus is indicated on the plans. All the air is recirculated, and humidity is supplied by the installation of water pans in the plenum chamber.

Typical Central System

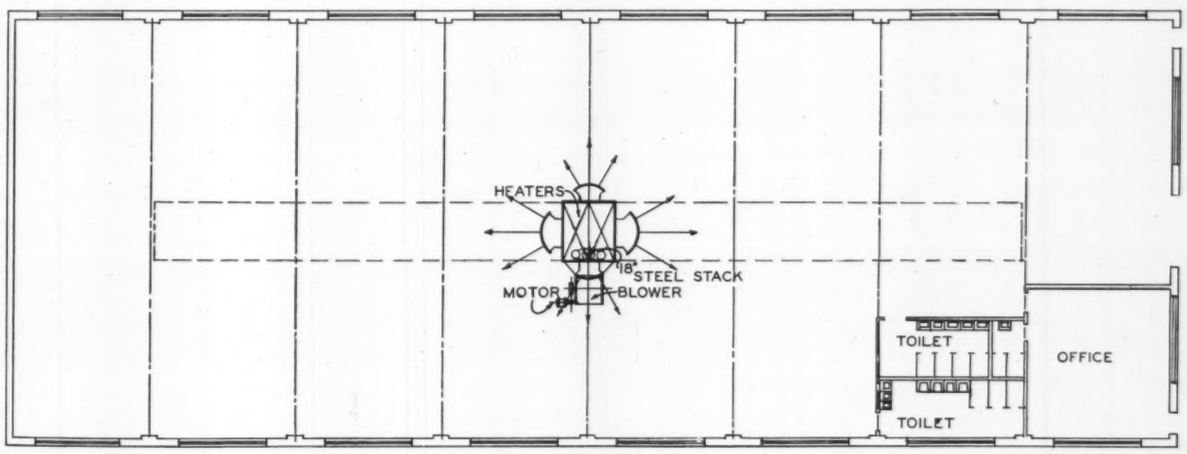
Fig. 3 is the central system with the warmed air piped to the various sections. One central main duct is shown, and an alternate two main system indicated by the dotted lines. The direct fired heaters are shown inside the building but may be installed in a separate room as shown in Fig. 1. Here the main duct and outlets are sized from the table of equivalents of round pipe and branches for equal friction per lineal foot. The desired resistance of the piping system is primarily decided. This may vary from 0.3 to 0.5 in. From the chart for determining resistance of air in ducts (Fig. 4), we establish the size of our main trunk duct and refer to the table for our outlets. The number of outlets will depend upon the spacing.

The main shown in Fig. 3 is 36 in. in diameter, and referring to Table 1, we find this is equal to fifteen and a half 12-in. outlets. We arrange sixteen outlets as shown, and again referring to the table we find that two 12-in. ducts are equal to one duct 16 in. in diameter approximately, and we have the main size at a point directly behind the two first discharges. Four 12-in. outlets are equal to a 21 in., as it falls between 20 in. and 22 in. approximately. The remainder of the main is sized in the same manner. Our main carries 16,700 c.f.m. or 1,043 c.f.m. approximately at each outlet. No doubt the outlets at the 36-in. size will discharge slightly more than 1,043 c.f.m., and those at the far end will fall as low as 950 c.f.m., but the table is practical.

The branch or outlet should leave the main at an angle of not more than 45 degrees, and the length and elbows in each outlet should be uniform. The end of the outlet may be flared to give the proper velocity. No return ducts are necessary and the movement of the return air will be noticeable only at a point within 20 ft. of the fan inlet. Provision should be made to take in some outside air at a point near the fan intake. This outside air will compensate for the leakage loss and tends to set up a slight pressure in the building. It is better to have the infiltration near the fan through an opening provided for it, than to have it scattered throughout the building. Where buffing or grinding machines are exhausting air to the outside, this outside air supply is important.

Ventilation fans have certain allowable velocities and speeds, but in the industrial heating system where noise is not objectionable, smaller high speed fans may be used. Outlet velocities up to 2,500 ft. at the fan discharge are allowable in this type of installation. Resistances in the direct fired system are generally low, an overall resistance of ½ to ¾ in. being the usual maximum. The problem of large ducts and less power, or smaller ducts, higher velocities, and more efficiency is debatable, and the writer will leave it to his readers.

To many designers, the ducts shown will seem too large, but 0.3 of an inch for the duct work as here outlined with velocities varying from 2,400 ft. at the near end to 1,490 ft. at the extreme end of the main are generally used for direct fired systems of fan blast heating in factory or industrial systems.



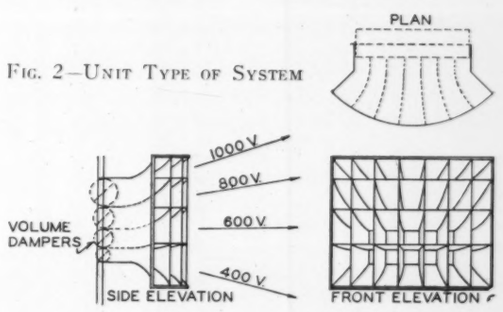


FIG. 2—UNIT TYPE OF SYSTEM

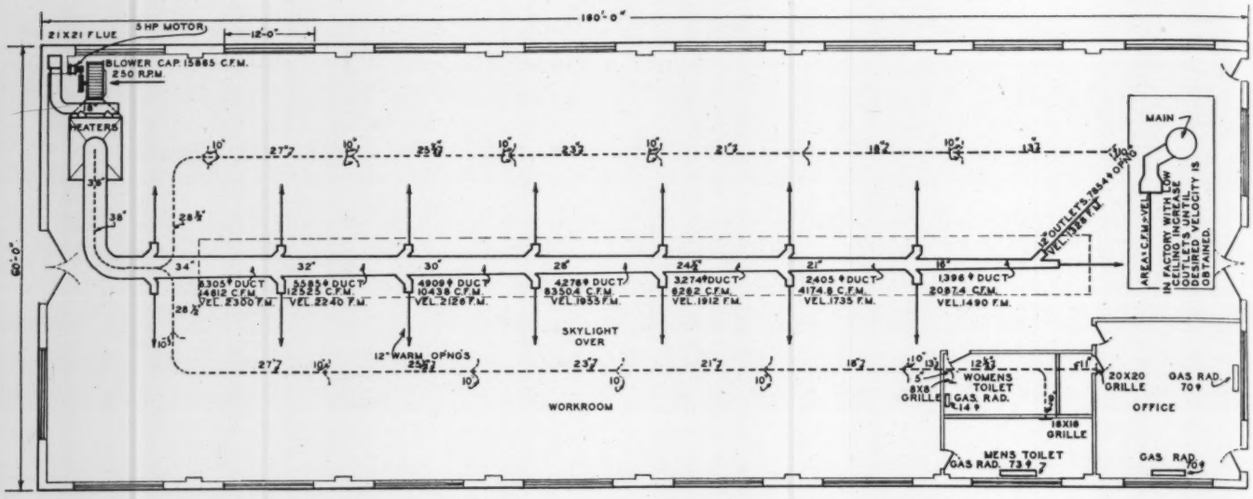

180'-0"
21X21 FLUE
5 HP MOTOR
12'-0"
BLOWER CAP. 15885 C.F.M.
250 R.P.M.
HEATERS
60'-0"
MAIN
AREA-C.F.M.-VEL.
IN FACTORY WITH LOW CEILINGS, SIZE OF OUTLETS UNTIL DESIRED VELOCITY IS OBTAINED.
8305 # DUCT 14612 C.F.M. VEL. 2300 F.M.
35854 DUCT 12525 C.F.M. VEL. 2240 F.M.
49098 DUCT 10438 C.F.M. VEL. 2126 F.M.
42789 DUCT 83504 C.F.M. VEL. 1955 F.M.
3274# DUCT 6262 C.F.M. VEL. 1912 F.M.
2405 # DUCT 4174.8 C.F.M. VEL. 1735 F.M.
1396 # DUCT 2087.4 C.F.M. VEL. 1490 F.M.
SKYLIGHT OVER
12" WARM OPNGS
WORKROOM
WOMENS TOILET GAS. RAD. 14 #
8X8 GRILLE
MENS TOILET GAS RAD. 73 # 7
20X20 GRILLE
GAS RAD. 70 #
OFFICE
18X18 GRILLE
GAS RAD. 70 #

FIG. 3—FLOOR PLAN. ALTERNATE PIPING LAY-OUT SHOWN IN DOTTED LINES

Heater Size

A practical rule for the determining of heater size requirements is based on required grate area. Attention is called to the well known fact that grate area alone does not constitute a safe factor, or a rigid one for determining the required heater or heaters. Inferior heaters might have large grate areas which would tend to confuse the designer or engineer. The ratio of prime heating surface to grate surface must be considered. Without any prepared data for this item, we must fall back on the old rule, that 20 to 1 is a minimum, and 40 to 1 is a maximum, with 30 to 35 to 1 a practical medium.

Combustion Rate

A combustion rate of 12 to 13 pounds of coal per sq. ft. of grate for direct fired hot blast systems is permissible. With heaters of this type, we base our grate area on the pounds of coal per hour required divided by the combustion rate. The pounds of coal per hour we determine from the formula:

Total c.f.m. $\times$ 60 $\times$ 0.24 $\times$ 0.068 $\times$ temp. rise fan intake to final temp., divided by 7,200.

Direct fired units are very flexible, and with the proper draft, combustion rates as high as 20 pounds of coal per sq. ft. of grate may be maintained; this, of course, with

TABLE 1—EQUIVALENT OF ROUND PIPE AND BRANCHES FOR EQUAL FRICTION PER LINEAL FOOT

DIAMETER INCHES	1 IN.	2 INS.	3 INS.	4 INS.	5 INS.	6 INS.	7 INS.	8 INS.	9 INS.	10 INS.	12 INS.	14 INS.	16 INS.	18 INS.	20 INS.	24 INS.
2	5.7	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3	15.6	2.8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
4	32.0	5.7	2.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
5	65.9	9.9	3.6	1.7	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
6	88.2	15.6	5.7	2.8	1.6	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
7	130	22.9	8.3	4.1	2.3	1.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
8	181	32.0	11.7	5.7	3.2	2.1	1.4	.....	.....	.....	.....	.....	.....	.....	.....	.....
9	243	43.0	15.6	7.6	4.3	2.8	1.9	1.3	.....	.....	.....	.....	.....	.....	.....	.....
10	316	55.9	20.3	9.9	5.7	3.6	2.4	1.7	1.3	.....	.....	.....	.....	.....	.....	.....
11	401	70.9	25.7	12.5	7.2	4.6	3.1	2.2	1.7	1.3	.....	.....	.....	.....	.....	.....
12	499	88.2	32.0	15.6	8.9	5.7	3.8	2.8	2.1	1.6	.....	.....	.....	.....	.....	.....
13	609	108	39.1	19.0	10.9	7.1	4.7	3.4	2.5	1.9	1.2	.....	.....	.....	.....	.....
14	733	130	47.0	22.9	13.1	8.3	5.7	4.1	3.0	2.3	1.5	.....	.....	.....	.....	.....
15	871	154	55.9	27.2	15.6	9.9	6.7	4.8	3.6	2.6	1.7	1.2	.....	.....	.....	.....
16	.....	181	65.7	32.0	18.3	11.7	7.9	5.7	4.2	3.2	2.1	1.4	.....	.....	.....	.....
17	.....	211	76.4	37.2	21.3	13.5	9.2	6.6	4.9	3.8	2.4	1.6	1.2	.....	.....	.....
18	.....	243	88.2	43.0	24.6	15.6	10.6	7.6	5.7	4.3	2.8	1.9	1.3	.....	.....	.....
19	.....	278	101	49.1	28.1	17.8	12.1	8.7	6.5	5.0	3.2	2.1	1.5	1.1	.....	.....
20	.....	316	115	55.9	32.0	20.3	13.8	9.9	7.4	5.7	3.6	2.4	1.7	1.3	.....	.....
22	.....	401	146	70.9	40.0	25.7	17.5	12.5	9.3	7.2	4.6	3.1	2.2	1.7	1.3	.....
24	.....	499	181	88.2	50.5	32.0	21.8	15.6	11.6	8.9	5.7	3.8	2.8	2.1	1.6	1
26	.....	609	221	108	61.7	39.1	26.6	19.0	14.2	10.9	7.1	4.7	3.4	2.5	1.9	1.2
28	.....	733	266	130	74.2	47.0	32.0	22.9	17.1	13.1	8.3	5.7	4.1	3.0	2.3	1.5
30	.....	871	316	154	88.2	55.9	38.0	27.2	20.3	15.6	9.9	6.7	4.8	3.6	2.8	1.7
36	.....	.....	499	243	130	88.2	60.0	43.0	32.0	24.6	15.6	10.6	7.6	5.7	4.3	2.8
42	.....	.....	733	357	205	130	88.2	63.2	47.0	36.2	24.6	15.6	11.2	8.3	6.4	4.1
48	.....	.....	.....	499	286	181	123	88.2	62.7	50.2	32.0	21.8	15.6	11.6	8.9	5.7
54	.....	.....	.....	.....	383	243	165	118	88.2	67.8	43.0	29.2	20.9	15.6	12.0	7.6
60	.....	.....	.....	.....	.....	316	215	154	115	88.2	55.9	38.0	27.2	20.3	15.6	9.9

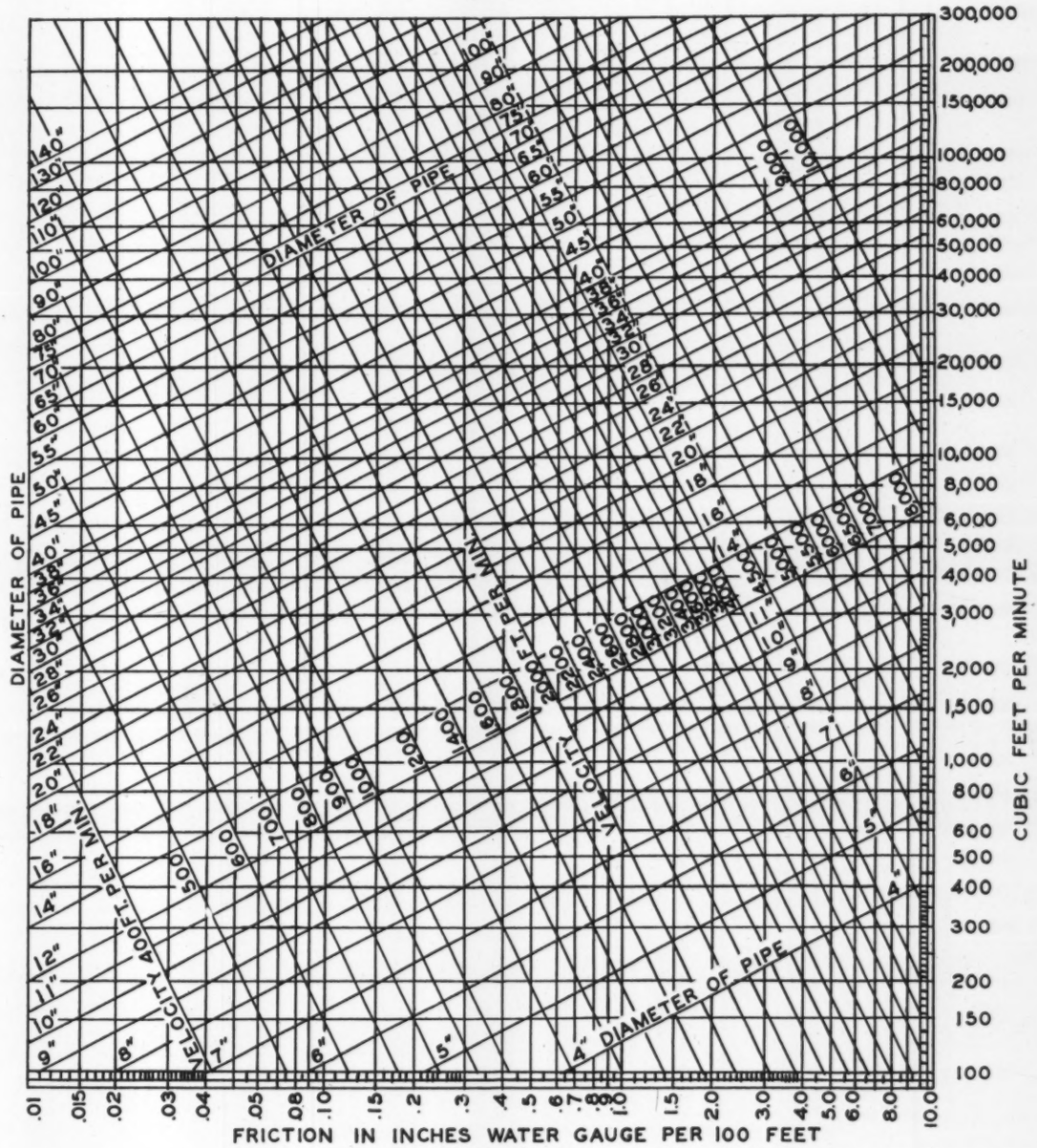

DIAMETER OF PIPE
VELOCITY 400 FT. PER MIN.
VELOCITY 1200 FT. PER MIN.
DIAMETER OF PIPE
CUBIC FEET PER MINUTE
FRICTION IN INCHES WATER GAUGE PER 100 FEET

FIG. 4—CHART FOR DETERMINING RESISTANCE OF AIR IN DUCTS

a drop in efficiency, and the above rule may be tempered with reason as it is based on the peak load. This peak load may not be in evidence for more than 3 or 4 days a heating season.

If a given installation required 18 sq. ft. of grate area for the peak load at a combustion rate of 12 pounds of coal per sq. ft., two heaters with 8 sq. ft. each might be specified if the next sized heater ran to 10 sq. ft. of grate. The combustion rate would, of course, be raised at the time the peak load was required. If the installation were in a northern section of the country, where the peak load might cover a period of weeks, the full 18 sq. ft. would be practical with the two 10 ft. heaters favored if there were no 9 ft. heaters obtainable.

With the fixed combustion rate, all direct fired heaters are, or should be, rated not on the B.tu basis, or equivalent sq. ft. of radiation, but on a given c.f.m. and temperature rise, in the writer's opinion. Here again available data are limited. The only tests known to the writer were based on a fixed combustion rate, and the c.f.m. varied from 5,500 to 10,000. The high point on the efficiency curve was reached at 8,500 c.f.m., proving that there is a direct relationship between the velocities over the heated surfaces and the combustion rate. Data relative to high and low combustion rates in relation to efficiency are not available. However, it is obvious that too little or too much air can be forced over direct fired heating surfaces. It is hoped that this valuable information will be forthcoming as this important branch of the heating and ventilating industry progresses.

A device that will control the combustion rate on direct fired heaters where coal is used as fuel is now used. This can function from a thermostat in the warm air plenum, as many firemen are prone to shovel on fresh fuel when the entire temperature regulation system is calling for cooler air. With an overheated plenum chamber, the overall thermal efficiency of the system is materially lowered.

Casing-in

The question of a practical solution is again encountered in the casing-in of direct fired heaters. Sheet steel casings are neat in appearance, and are easy to remove to get at the heaters, but must be built to order. Brick casings are practical, and if building conditions are unfavorable, or dimension lines are not true, brick casing walls may be varied to meet the conditions.

Where the heaters are confined to a small furnace room as shown on Fig. 1, the blower must be placed in a separate room, or the smoke flue draft will be materially affected. Space in front of the large return grille may be used for the seasoning of material. In the forementioned auto-body works, this space is utilized for the drying and seasoning of the timber that goes into the framework of cars and trucks.

A Storage Plant Refrigeration System

By E. M. Mittendorff

A STORAGE company in southern Illinois recently built a three-story building for the storage of eggs of all kinds.

Experience had shown that the eggs should be kept in a room at 30 F air temperature and 80 per cent relative humidity.

Fig. 1 shows the general plan for each floor. Brine at 10 F is supplied to the building by a 3 in. pipe from ammonia compressors located in the power house about 150 feet away. This brine goes mainly to two refrigeration units on each floor from which air at 30 F and 80 per cent relative humidity is supplied to the room and recirculated. Close to the ceiling runs a 2-in. pipe which is perforated on its upper circumference and serves to expel the odor in a manner to be explained later, when natural flow is impossible. In the center of the room are two electric temperature control switches, one controlling the fan motor while the other controls the brine spray pump motor.

Fig. 2 shows the general construction of each refrigeration unit. As long as the temperature of the room is below 30 F, the electric control switch in the room will be closed and the fan *M* will circulate the air. Recirculated air from the room enters the unit through the grille *C*. It then is sucked past the cold brine coils *E* where it is chilled to 25 F. To keep this temperature as accurate as possible, a direct-acting temperature regulator *G* with a piston type reverse-acting valve is installed which controls the brine flow so as to keep the air temperature at the desired point (25 F). With that temperature, the rising air is saturated to practically 100 per cent relative humidity by the brine sprays escaping through the nozzles *N*. The fan finally blows the air into the room through the openings *D*. At its exit from the unit, the air temperature has again risen to about 30 F and has the desired 80 per cent relative humidity.

The above operation takes place when the air temperature has to be held down, such as will be the case during the summer months. The unit is equipped with a warm brine coil

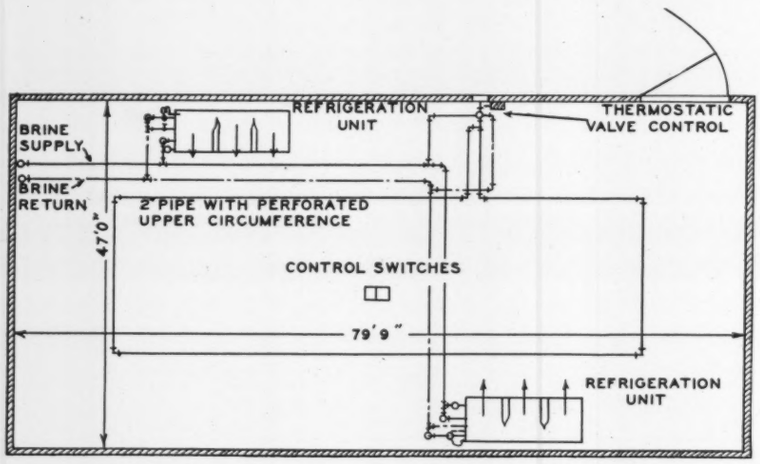


FIG. 1—GENERAL PLAN OF EACH FLOOR

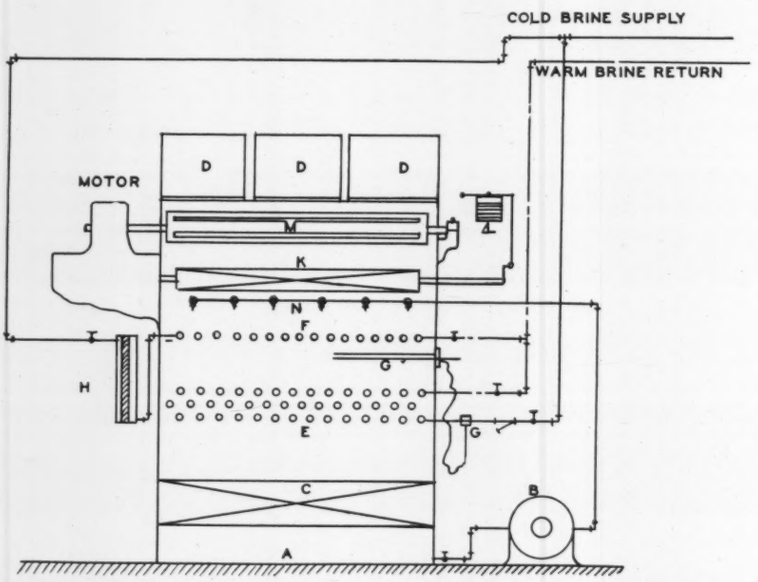


FIG. 2—GENERAL CONSTRUCTION OF EACH REFRIGERATION UNIT. *A*—BRINE TANK. *B*—BRINE PUMP. *C*—WARM AIR INLET. *D*—COLD AIR OUTLET. *E*—COLD BRINE COIL. *F*—WARM BRINE COIL. *G*—TEMP. REGULATOR. *H*—ELECTRIC BRINE HEATING ELEMENT. *K*—RECIRCULATING DAMPER. *L*—AUTOMATIC DAMPER REGULATOR. *M*—CIRCULATING FAN. *N*—BRINE SPRAY NOZZLES

for severe winter days when it may be necessary to raise the air temperature. The brine is warmed by means of an electric wire coil *H* which is inside a pipe. Both the brine valves, as well as the electric current switch for this equipment, are hand-controlled as there are only a few days in the year when it is necessary to use them. When the room temperature, due to the outside temperature, drops slightly below 30 F, the recirculating damper *K*, operated by the automatic thermostatic control *L*, will open and recirculate the air.

The removal of the odor and warm air which will collect under the ceiling of the room is taken care of in an

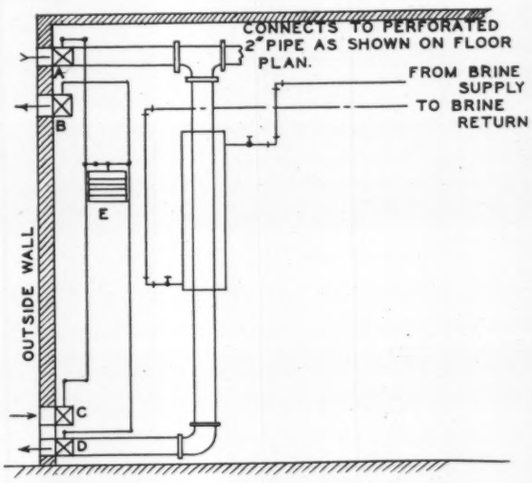


FIG. 3—DETAIL OF ODOR REMOVING MECHANISM

unusual manner. All around the room, under the ceiling and about five feet from the walls, is a 2-in. pipe which is perforated on its upper circumference. During days when natural passage of warm air and odor is possible to the outside, the thermostatic valve control *E*, in Fig. 3, opens valve *B*, closes *D* and *A* and also opens *C*. In that way the odor under the ceiling will pass out of valve and opening in wall *B* and be replaced by fresh air coming in through opening and valve *C*. When the atmospheric temperature is such that no natural exchange of air is possible, valves *C* and *B* will close and *A* and *D* will open. Air and odor are filling the 2-in. perforated pipe under the ceiling and circulation is started by sending brine around a vertical branch of this 2-in. pipe. This cools the foul air and it will sink down the pipe and out into the atmosphere through valve *D*. Replacement will take place through valve *A*.

The Accuracy of the Orifice Meter

QUESTION:—"Although the orifice meter has been used extensively as a standard method of measurement for large volumes, some believe that it is not as accurate as other methods and that it is a new method of measurement. I would like to have your opinion on this.

"The Chaldeans measured distances in the heavens by taking the width of the sun's disc as a unit. By comparing the quantity of water discharged through an orifice in a jar in the time occupied by the sun crossing the horizon on the morning of the equinox with the amount discharged through the same orifice by the next sunrise, they discovered that the amount discharged between the two risings of the sun was seven hundred and twenty times the amount discharged during sunrise on the equinoctial morning. They thus inferred that the sun's orbit measured seven hundred and twenty times his disc, and from this they derived a unit to measure space and time.

"The Chaldeans discovered and recorded the fact that each cycle of the moon's eclipses is completed in a period of two hundred and twenty-three months, and from this discovery they computed the length of the synodic and periodic months so accurately that modern astronomers have found the calculation to fall short less than five seconds of our time. They carefully recorded all the results of their observations. The Greek Callisthenes, who had accompanied the expedition of Alexander the Great, sent to Aristotle, from Babylon, a series of tablets on which were inscriptions recording astronomical observations dating as far back as 1903 years before the year 331 B. C., the year that Alexander entered that city.

"This would appear to show that the orifice principle was used many years ago and that is as accurate for large volumes as time itself."—*From a Subscriber.*

Prof. A. A. Cole, Purdue University, Lafayette, Ind., in commenting upon the above, says:

The flow of any homogeneous substance through an orifice will always obey simple laws of nature. Engineers are more especially interested in the laws governing the flow of liquids, vapors, and gases through orifices. The physical laws of flow are invariable.

The ability of engineers to use such physical laws to assist them in measuring flow depend upon several factors, including a thorough knowledge of the laws of flow, and skill in constructing and operating apparatus in conformation to these laws.

Skill Necessary

Research in the laboratory reveals the laws and teaches engineers how to predict results through the application of natural phenomena. Skill must be of two kinds; skill in construction, and skill in operation. After research has revealed the laws, an instrument must be devised by which the natural changes may be measured. Such measurements in the present instance will involve changes in pressure, velocity and heat change.

With a proper instrument at hand, it must be operated and cared for by an interested and skillful operator. Success depends very largely upon the operator, for no instrument can function properly unless it is properly adjusted.

When all of these requisites of knowledge and skill have been fulfilled the accuracy of an orifice meter can not be questioned. The accuracy of such a meter is of a high order. In practice, an orifice meter should give values within a two per cent limit of accuracy.

The principle of the orifice meter is not new although its application to power plant use is comparatively modern. As soon as the power plant outgrew the ability of scales to weigh coal and water, the orifice meter came to the front. With boilers over 200 horsepower it is impossible to operate without an orifice meter. Undoubtedly the future will see a great increase in the use of the orifice meter.

Underground Piping for District Heating

By A. A. Sellke and G. D. Winans

TO design an underground steam distribution system for central station heating properly requires a study of several factors, among which are present load, expected growth, sub-surface conditions and pressure requirements. Future expansion of an underground system must be considered in its design, but care must be taken to avoid large expenditures for future business.

There are two methods of underground steam distribution, namely, the trunk main and the feeder method. The trunk main method consists of a main from the heating plant to the district served, with decreasing pipe sizes as the quantity of steam fed decreases. Lateral pipes are connected to the trunk main at street intersections.

The feeder system consists of feeders which extend from the heating plant to strategical points on the heating system. Pressures are generally controlled at these points by means of long distance gages.

Increased demands on an overloaded trunk main system may be supplied by the installation of feeder pipes.

The type of underground construction to be used is governed by territory served and ground conditions. In a purely residential district, which has slight prospects of becoming a business section, cheaper types of construction may be used. A district which may become flooded frequently on account of inadequate sewers or tidewater will require quite an elaborate and costly construction, such as cast iron conduit with caulked joints.

In the congested downtown districts of the larger cities, the protection of underground steam piping is essential. A great deal of excavation is going on at times which will bare the conduit. Therefore it is necessary to have a form of construction which has sufficient mechanical strength to withstand the abuses of excavation both below and above the steam line conduit.

The obstructions in streets will determine the depth and position of the mains. Knowledge of the earth conditions is necessary to determine the methods of trenching.

A thorough acquaintance with the methods of construction and installation as used by other utilities in the locality in question enables one to tell approximately at what depth and location these possible obstructions may be found. If the soil is clay and has not been previously disturbed, very little bracing is necessary. If this happens to be the condition, and the engineer in charge has the acquaintance mentioned, then a trenching machine will help materially in cutting the cost of the excavating. In sandy or wet soil, a trenching machine cannot be used. Here the bracing should be installed, bearing in mind that pipes of certain lengths will have to be lowered into the ditch with as little disturbance to the bracing as possible. In the downtown sections of any large city, it is practically impossible to use a trencher due to traffic conditions which might necessitate the work being done at night, or else the obstructions are too frequent and the danger of damaging these is too great to permit its use.

Backfilling and resurfacing are usually done under city supervision and according to city specifications. The length of time during which the responsibility of the backfill or pavement rests with the utility is also specified, and is usually five years.

Capacity of Steam Pipes

The carrying capacity of a steam pipe is a function of the pressure drop which may be allowed along the length of the pipe. It is possible to feed as much steam through a small pipe with a large pressure drop as it is through a large pipe with a small drop. This can be seen by referring to Fig. 1, which is a graphical solution of Unwin's formula. Velocity should not be considered as a factor in the design of underground steam mains. Velocities of 75,000 feet per minute have been observed in feeders without vibration or erosion, and velocities of 50,000 feet per minute are common.

Example: It is desired to feed 100,000 pounds of steam per hour through a 12-inch pipe, 1,000 feet long. Determine the final pressure.

It will be noted that the pressure drop in the chart is given for lengths of 100 feet. Therefore it is necessary to estimate what the final pressure will be to determine the average pressure. This average pressure is then used as the initial pressure for lengths greater than 100 feet. With some practice, it will be possible to tell quite accurately what drops certain flows will cause. However, it is often necessary to make several assumptions for the final pressure before obtaining the correct result.

From the "Quantity of Steam" line for nominal pipe diameter of 12 inches, find a flow of 100,000 pounds per hour. Draw a vertical line to a pressure line somewhat below 100 pounds gage, say 90 pounds. It will be seen that the pressure drop will be slightly more than 0.7 pounds per 100 feet, or 7 pounds for 1,000 feet. Now let us assume the final pressure as 92 pounds gage. Therefore, this will give an average pressure of 96 pounds.

From the "Quantity of Steam" line for nominal pipe diameter of 12 inches at a flow of 100,000 pounds per hour, draw a vertical line to the pressure line for 96 pounds gage; where the vertical line intersects the flow line may be read the pressure drop as 0.73 per 100 feet, or 7.3 pounds per 1,000 feet.

The final pressure is $100 - 7.3 = 92.7$ pounds. If

greater accuracy is desired, the problem is reworked using the new final pressure of 92.7 pounds.

Pressure necessary at the customer's service depends on the purpose for which steam is used. If used for

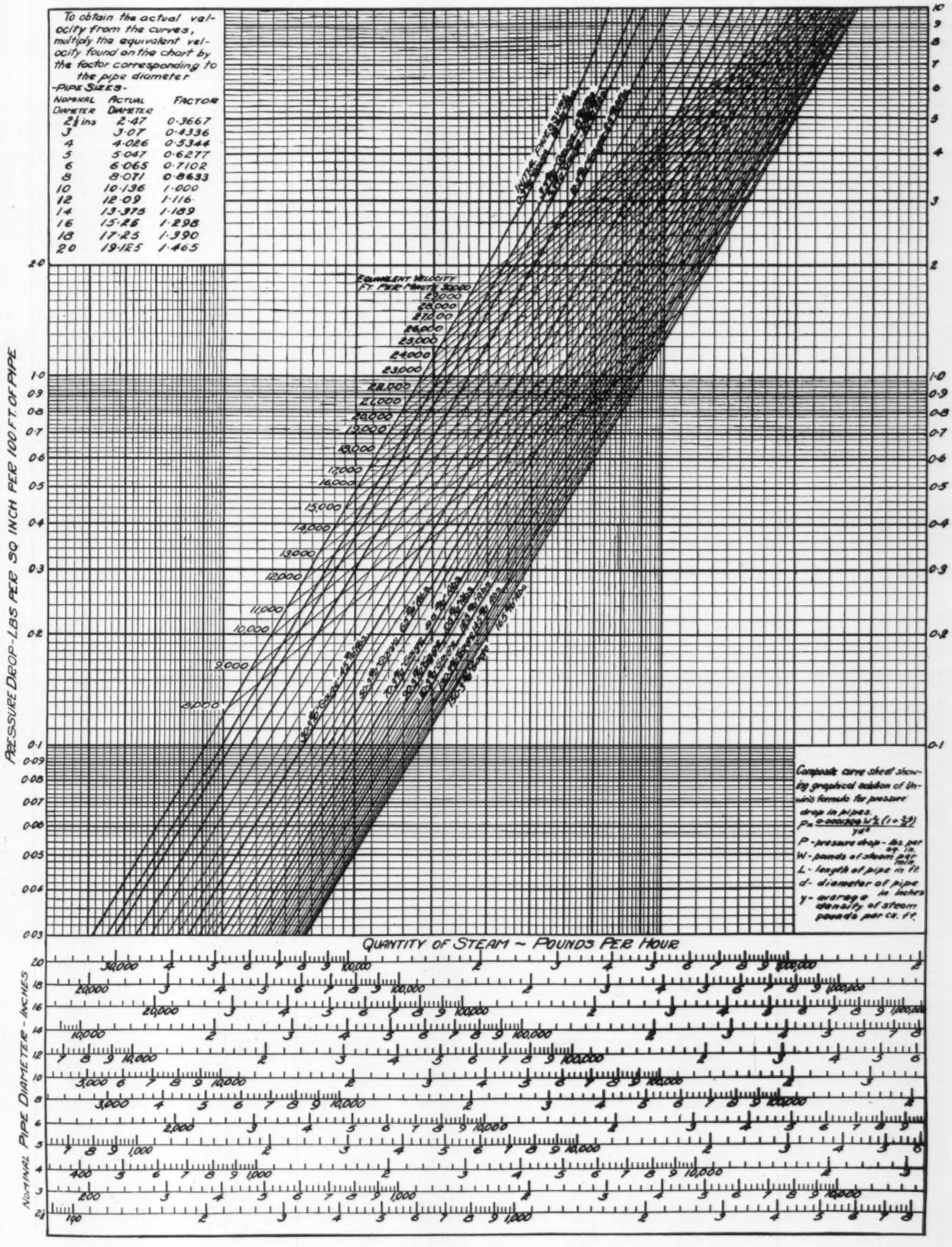

To obtain the actual velocity from the curves, multiply the equivalent velocity found on the chart by the factor corresponding to the pipe diameter
-PIPE SIZES-
NOMINAL DIAMETER | ACTUAL DIAMETER | FACTOR
2½ ins | 2·47 | 0·3667
3 | 3·07 | 0·4336
4 | 4·026 | 0·5344
5 | 5·047 | 0·6277
6 | 6·065 | 0·7102
8 | 8·071 | 0·8633
10 | 10·136 | 1·000
12 | 12·09 | 1·116
14 | 13·375 | 1·189
16 | 15·26 | 1·298
18 | 17·25 | 1·390
20 | 19·125 | 1·465
EQUIVALENT VELOCITY FT. PER MINUTE
PRESSURE DROP-LBS PER SQ INCH PER 100 FT. OF PIPE
QUANTITY OF STEAM ~ POUNDS PER HOUR
NOMINAL PIPE DIAMETER - INCHES
Composite curve sheet showing graphical solution of Unwin's formula for pressure drop in pipes.
P - pressure drop - lbs per sq in.
W - pounds of steam per min.
L - length of pipe in ft
d - diameter of pipe in inches
γ - average density of steam pounds per cu. ft.

FIG. 1—COMPOSITE CURVE SHEET SHOWING GRAPHICAL SOLUTION OF UNWIN'S FORMULA FOR PRESSURE DROP IN PIPES

TABLE 1—OUTLINE OF PIPING STANDARDS OF ONE COMPANY FOR UNDERGROUND STEAM PIPE CONSTRUCTION. MAINS AND SERVICES ARE OPERATED AT ABOUT 35 LB. PER SQ. IN. PRESSURE. FEEDERS ARE OPERATED AT FROM 35 TO 125 LB. PRESSURE.

LINE		SURFACE MAINS	SURFACE FEEDERS	TUNNEL MAINS	TUNNEL FEEDERS	SERVICES AND DRIP CONNECTIONS
Weight of valves, flanges and fittings		125 lbs. W. S. P.	250 lbs. W. S. P.	250 lbs. W. S. P.	250 lbs. W. S. P.	Screwed fittings, 250 lbs. W. S. P., flanged fittings and valves, 125 lbs. W. S. P.
Pipe		Standard weight lap welded steel	Standard weight lap welded steel	Standard weight lap welded steel	Standard weight lap welded steel	Standard weight lap welded steel.
Joint at fittings	Size	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	Maximum 4 inches... Minimum 6 inches.
	Style	Std. companion flanges, 125 lb. W. S. P. forged steel	Square lap van stone 250 lb. W. S. P. forged steel	Square lap van stone steel flanges, 250 W. S. P.	Square lap van stone steel flanges, 250 W. S. P.	Screwed... C. I. screwed flange.
	Gasket	Asbestos composition	Asbestos composition	Asbestos composition	Asbestos composition	Asbestos composition
Joints between fittings	Size	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	
	Style	Welded	Welded	Square lap, van stone 250 lb. W. S. P. steel (or C. I.), flanged	Square lap, van stone, 250 lb. W. S. P. forged steel, flanged	Welded at main or flanged.
	Gasket	None	None	Asbestos composition	Asbestos composition	Asbestos composition
Screwed valves	Maximum size					3 inches... Over 3 inches, see "flanged valves."
	Material					Up to 2 inches, brass 2½ and 3-in. cast iron
Flanged valves	Minimum size	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches... Minimum 10 inches.	Minimum 4 inches... Minimum 10 inches.	4 inches
	Type material	125 lbs. W. S. P. solid wedge gate valve, non-rising stem	250 lbs. W. S. P. solid wedge gate valve, non-rising stem, I. B. B. M.	250 lbs. W. S. P. solid wedge gate valve, non-rising stem, / 250 lbs. W. S. P. solid wedge gate valve, non-rising stem, steel body, brass mounted	250 lbs. W. S. P. solid wedge gate valve, non-rising stem, I. B. B. M. / 250 lbs. W. S. P. solid wedge gate valve, non-rising stem, steel body, brass mounted	Same as surface mains.
Expansion joints	Size	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	Minimum 4 inches	
	Style	Slip joint, 125 lbs. W. S. P.	Slip joint, 250 lbs. W. S. P.	Slip joint 250 lbs. W. S. P.	Slip joint 250 lbs. W. S. P.	
Traps	Size					
	Style	High press. float type, except low press. in Willis District	Same as surface mains.	Same as surface mains	Same as surface mains	Same as surface mains.
Screwed fittings	Maximum size					3 inches.
	Material					250 lbs. W. S. P. cast iron.
Flanged fittings	Minimum size	4 inches	4 inches	4 inches... 10 inches	4 inches... 10 inches	4 inches.
	Material	Cast iron	Cast iron	Cast iron... Cast steel	Cast iron... Cast steel	Cast iron.
Miscellaneous items	Wedges	Cast iron	Cast iron	Cast iron	Cast iron	Cast iron.
	Line anchor	Cast steel	Cast steel			Cast steel.
	Brackets	Cast iron	Cast iron			Cast iron.
	Rollers	Cast iron	Cast iron	Cast iron	Cast iron	Cast iron.
Insulation	Pipe	Laminated asbestos sponge, felted with waterproofing jacket attached. Covering for pipe 10 inches and smaller, 1 inch thick. Covering for pipe 12 inches and larger, 1½ inch thick	Same as surface mains.	3-inch 85% magnesia 10 oz. canvas	3-inch 85% magnesia 10 oz. canvas	Same as surface mains.
	Fittings	1½ inches of asbestos cement. Block magnesia on large fittings in manholes use ½ portland cement	Same as surface mains	3-inch 85% magnesia block	3-inch 85% magnesia block	Same as surface mains.

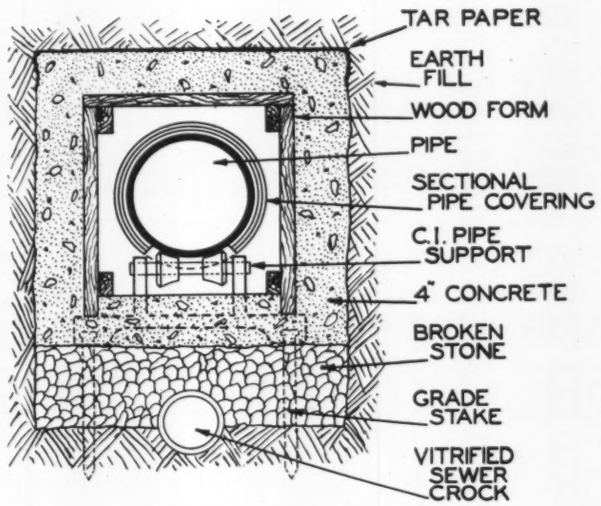


FIG. 2—CROSS-SECTION OF CONCRETE CONDUIT

heating only, the pressure may be lower, of course, than if the steam is used in manufacturing processes or for driving machinery.

Kinds of Pipe

The common materials used in underground piping for district heating systems are steel and wrought iron pipe encased in some sort of protective conduit. In a few cases, cast iron pipe has been used, but in general it is confined to return lines. The pipe encased in its conduit is buried to whatever depth is necessary to clear the existing street obstructions. One type of conduit is shown in Fig. 2. Costs of conduits installed vary greatly in different localities due to sub-surface conditions. In Detroit, the average cost of this type of construction is as follows:

Pipe Size	Approx. Cost per Ft. in Dollars
6″	18
8″	20
10″	22½
12″	25
16″	30

Drainage

One of the important factors in the successful operation of a buried steam line is its drainage, both internal and external. Almost every type of conduit is provided with some sort of underdrainage. Vapor may be present in the conduit, due to water from leaking water mains, foreign conduits or sewers and from steam leaks. Service thimbles or stuffing boxes as shown in Fig. 3, if placed on building services, will prevent vapor entering the building through the steam conduit.

Manholes are located where necessary to provide for the installation of valves, slip joints, drip traps and later main connections. Manholes must be placed at low

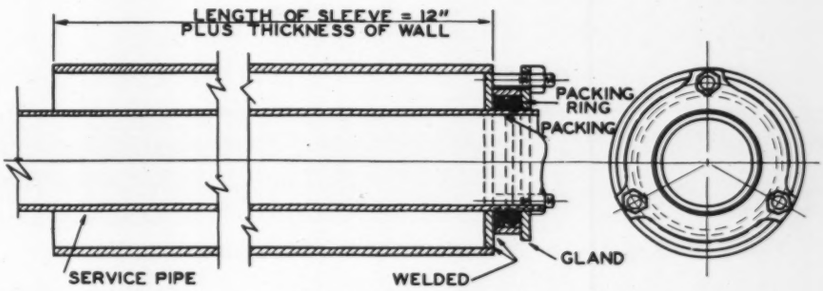
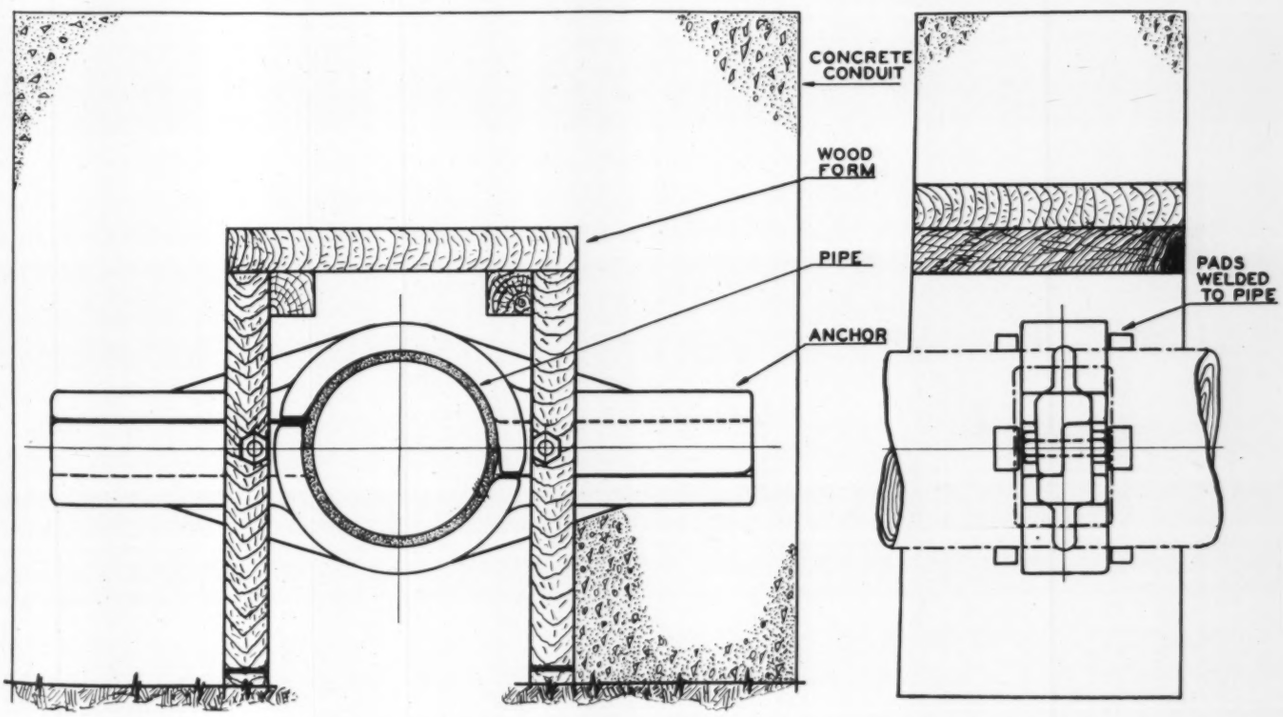


FIG. 3—SERVICE THIMBLE



ASSEMBLY OF STEAM PIPE ANCHORS

FIG. 4—CAST STEEL ANCHORS

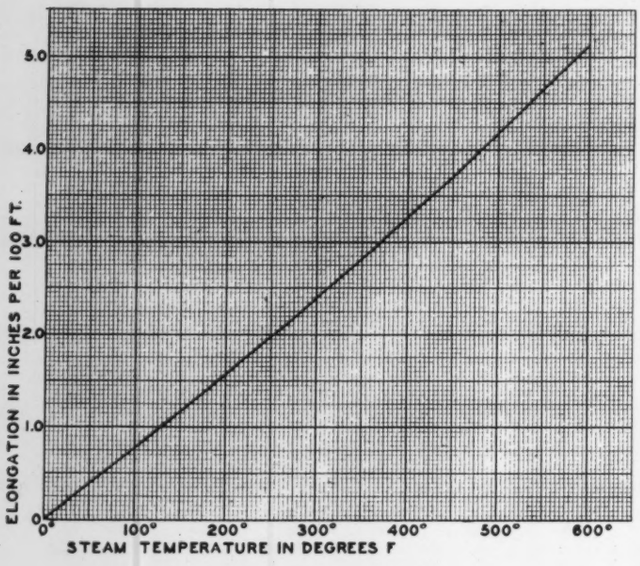


FIG. 5—ELONGATION OF STEEL PIPE COMPUTED BY HOLBORN & DAY'S FORMULA:

$$L_t = L_o\left[1+a\left(\frac{t-32}{1000}\right)+b\left(\frac{t-32}{1000}\right)^2\right]$$

where L_t is length at temperature t, L_o is length at 32 F. For steel, a=.006212 and b=.001623. Curve is for steel pipe only.

points of the mains to provide for drip trap connections.

It is desirable to design the system so that anchors are installed in the line at high points midway between the manholes and in the manholes at the slip joints. Details of one type of anchor are shown in Fig. 4. Where guides are deemed necessary, the same pair of castings as shown in Fig. 4 may be used. When used as guides, the pads welded to the pipe are omitted.

Fusion welding, both electric arc and oxy-acetylene, is being widely used with good results.

In most types of construction, pipe supports are installed every 12 or 15 feet. Fig. 4 shows details of pipe supports. The minimum pitch for steam piping should be 4 inches in 100 feet.

Insulation

One difference between underground steam piping and that for other utility service such as water and gas is the necessity of protecting it from heat losses. The insulation required depends on the pressure or temperature of the steam. In general, the covering should be capable of being submerged in water and regaining its original insulating qualities after being dried out. Loose filler insulations which fill the entire space between the pipe and conduit have been used with success, but the writers believe that the general trend is to sectional covering applied with a waterproof jacket on the outside. The latter type of insulation offers no resistance to the drainage of water which might enter the conduit. Heat losses from well-insulated pipes in dry soil vary from 50 to 75 B.tu. per hour per square foot of pipe surface.

Expansion

Expansion and contraction of piping are absorbed by some form of expansion joint, of which there are several designs: the slip, corrugated, and variator type. The latter two types of joints are packless and may be buried. The slip type joint must be placed in manholes to permit packing. The amount of expansion may be determined by means of Holburn and Day's formula. The graphical solution of this formula for steel pipe is shown in Fig. 5.

The length of pipe allowable for a single slip joint is governed by the number and location of service and lateral connections. In this type of construction (Fig. 2), provision should be made in the conduit to allow for expansion at the point where service pipe or lateral is connected to the main. Where maximum operating pressure are 50 pounds gage, it is safe to install one slip type expansion joint for 250 feet of pipe.

Service Piping

Service pipes are connected to the mains either by welded or screwed joints, being taken from the top of the main and generally pitched to the main. If necessary to grade the service pipe toward the building, the service must be dripped to prevent condensation from the service going to the customer's meter. It is customary for the heating company to take the condensation in the service line from the main to the customer's valve, inside of the building.

Several Factors Influence Transmission

The distance which steam may be transmitted economically depends entirely on local conditions, such as cost of land for plant sites in the vicinity of the load as compared with the cost of steam lines, coal handling facilities, character of the business served and plant load factor.

Welding Tanks

Welding should be completed without interruption, the tank being turned so that the welding is always done in a position convenient for the operator. When the end of the girth seam is reached, the operator should remelt and reweld into the previously welded portion and should heat about 12 in. of the first part of the weld to a dull red, continuing the heating until the end of the weld has cooled to a dull red. Before welding the inside vee, the bottom of the first vee is carefully chipped.

Shell nozzles are welded in somewhat the same way as a position weld in pipe. The shell is turned so the axis of the nozzle is horizontal. The weld is started at the bottom and continued up one side. The shell is then rotated 180 deg. and the weld continued.

Low Pressure Steam Characteristics and Steam Flow *and* Typical Calculations

By Prof. W. R. Woolrich *and* L. Holdredge

CONFUSION sometimes exists in the minds of many steam users on account of their failure to comprehend the relationships between steam pressures, temperatures and heat values. This confusion becomes the more marked in treating with low pressures when vacuum values are introduced. To most people, vacuum is merely something that is not there.

Unfortunately, the only indicator on the average heating unit is a steam pressure gage. The gage finger must stand for the measure of what is going out. On this basis plain arithmetic indicates that 4 lb. is twice as great as 2 lb.; therefore, it is twice as hot and twice as effective.

To visualize the true relation between pressure and temperature, Fig. 1 is given. The boiling point of any liquid rises and falls very rapidly with the change of pressure above the liquid surface. Since the heat transfer rate is dependent upon the difference of temperature inside and outside the radiator, this temperature rise indicates to some measure the increase of heat flow with the increase of steam pressure.

Fig. 1 was drawn showing actual temperatures plotted against gage pressures. To more closely approximate the true conditions, the curve of Fig. 2 is developed from Fig. 1. Fig. 2 shows the temperature difference between the inside and the outside of the radiator for low pressures, assuming the outside room temperature of 70 F.

In tabular form, some interesting comparisons can be made from Table 1.

It is common practice on heat radiation problems to assume that the average difference between inside and outside temperatures of a steam radiator will be 145 F.

These temperature differences are no indication whatever of the heat provided by each pound of steam circulated. They are only a factor in determining the amount of heat that will be transferred through the radiator wall.

Total Heat of Steam

The total heat in steam does not vary in any such ratios as the pressure and temperature characteristics. As discussed in previous articles, the total heat is made up of the "heat of the liquid" and the "latent heat of the steam." To re-define "heat of the liquid," it is the amount of heat in B.tu to raise the temperature of one pound of water from 32 F to the boiling temperature at the pressure being considered. The latent heat is the amount of heat necessary to change the pound of water from the condition of water at the boiling point

TABLE 1—BOILING POINTS AND TEMPERATURE DIFFERENCES IN 70 F ROOMS AT DIFFERENT PRESSURES

GAGE PRESSURE	BOILING POINT	TEMPERATURE DIFFERENCE BETWEEN INSIDE OF RADIATOR AND 70 F IN ROOM, IN F
28 in. Vacuum	101	31
25 in. Vacuum	134	64
20 in. Vacuum	161	91
15 in. Vacuum	179	109
10 in. Vacuum	192	122
5 in. Vacuum	203	133
0 in. Vacuum	212	142
5 lb.	227	157
10 lb.	239	169
15 lb.	250	180
25 lb.	267	197
50 lb.	298	228
100 lb.	338	268

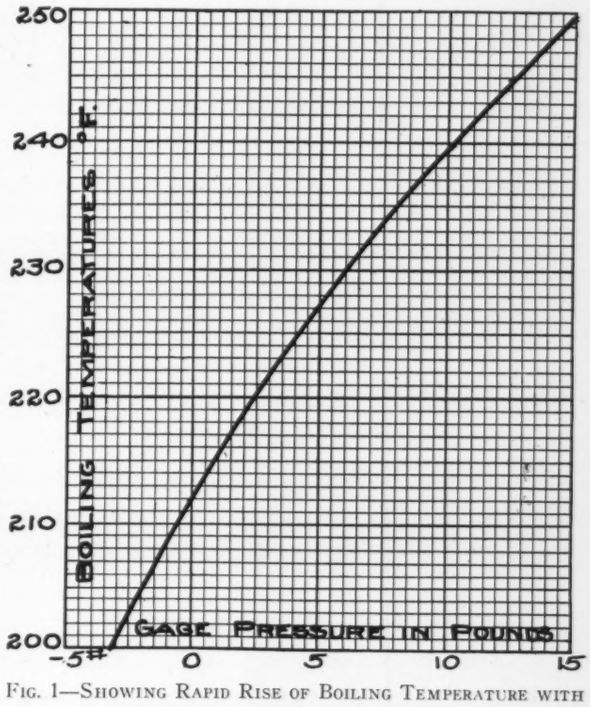


FIG. 1—SHOWING RAPID RISE OF BOILING TEMPERATURE WITH INCREASE OF GAGE PRESSURE

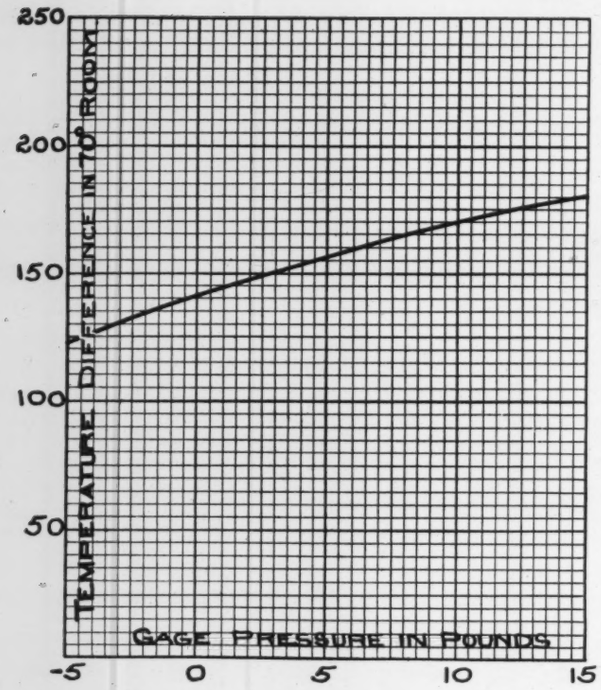


FIG. 2—TEMPERATURE DIFFERENCES INSIDE AND OUTSIDE A RADIATOR AT DIFFERENT GAGE PRESSURES AND A ROOM TEMPERATURE OF 70 F

to that of steam at the boiling point at the pressure considered.

Fig. 3 shows in curve form how slowly, relatively, the total heat changes with change of pressure. Table 2 gives some interesting comparisons of these values at different pressures.

Effect of Altitude on Boiling Points

Most steam users follow the assumption that the boiling point of water is 212 F. Under sea level conditions, the temperature of boiling water is 212 F when atmospheric pressures are standard, but at elevations higher than sea level the boiling temperature drops because the atmospheric pressure is reduced.

Fig. 4 shows the boiling temperatures at different altitudes under standard barometric conditions.

TABLE 2—HEAT OF LIQUID, LATENT HEAT AND TOTAL HEAT OF STEAM AT DIFFERENT PRESSURES IN B.TU

GAGE PRESSURE	HEAT OF LIQUID ABOVE 32 F	LATENT HEAT OF VAPORIZATION	TOTAL HEAT
28 in. Vacuum	69	1035	1104
25 in. Vacuum	102	1016	1118
20 in. Vacuum	129	1001	1130
15 in. Vacuum	147	990	1137
10 in. Vacuum	160	982	1142
5 in. Vacuum	171	976	1147
4 in. Vacuum	173	975	1148
3 in. Vacuum	175	973	1148
2 in. Vacuum	177	972	1149
1 in. Vacuum	178	977	1149
0 in. Vacuum	180	970	1150
1 lb.	183	969	1152
2 lb.	187	966	1153
3 lb.	190	964	1154
4 lb.	193	962	1155
5 lb.	195	961	1156
10 lb.	208	952	1160
15 lb.	218	946	1164
25 lb.	237	932	1169
50 lb.	267	911	1178
100 lb.	309	880	1189

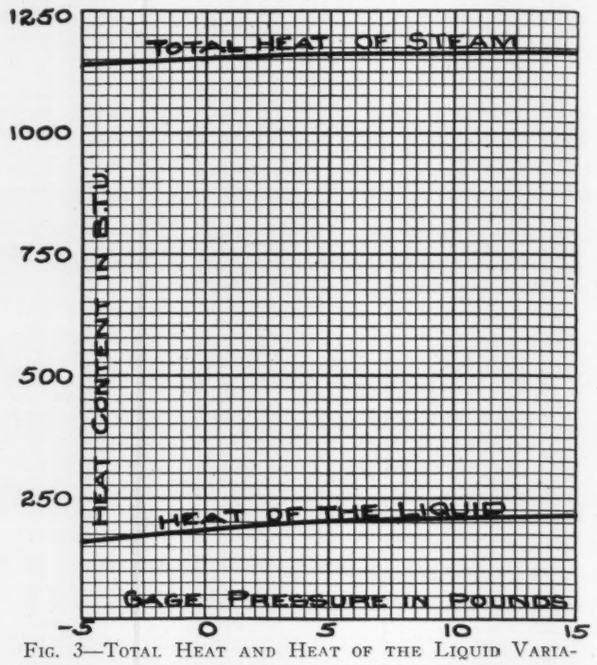


FIG. 3—TOTAL HEAT AND HEAT OF THE LIQUID VARIATIONS WITH CHANGE OF PRESSURE

Steam Quality

Technically, steam may be wet, dry or superheated. Practically, the possibility that it will ever be exactly 100 per cent dry steam is very remote, thus the two most common states of steam are wet and superheated.

Wet Steam: Steam leaving the heating boiler is usually wet. In the process of ebullition, the steam bubbles, breaking through the surface of the water, spray upwards myriad particles of water. The steam moving rapidly upward toward the boiler valves or outlets carries these particles with it.

Out in the heating system, all degrees of wet steam are encountered. Naturally, since at the radiator outlet the steam has become nearly 100 per cent water, and at the boiler we had nearly 100 per cent steam, we must expect to find, always, every possible degree of steam wetness in heating systems.

Superheating: Superheating is most commonly caused by (1) heating the steam up in some pipe or container apart from the presence of water, (2) by dropping the steam pressure through a pressure reducing valve or (3) by conducting the steam at a high velocity through a well-insulated steam main with a resultant pressure drop.

Superheating within the boiler in separate piping is not so very common in heating practice. Usually, where it is practiced, the superheating is done for power or process work and not for the heating load.

But superheating through pressure reducing valves and in pipe lines is more common to heating practice. An examination of Table 2 reveals that the total heat of low pressure steam is considerably less than that of

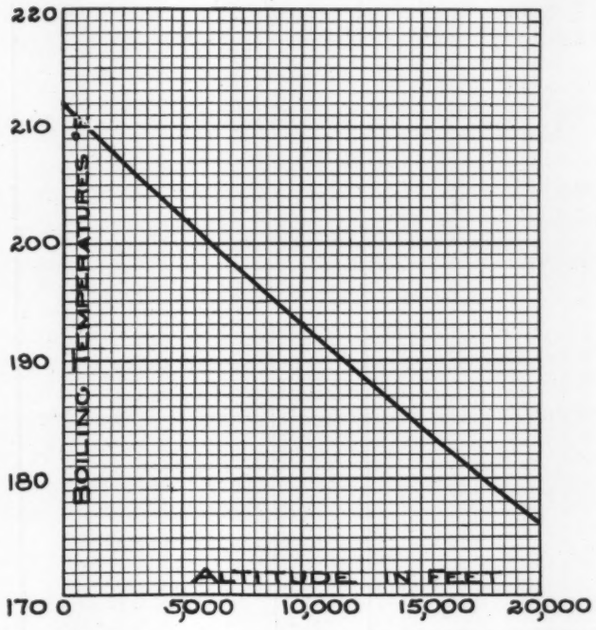


FIG. 4—CHANGE OF BOILING POINT WITH CHANGES OF ALTITUDE

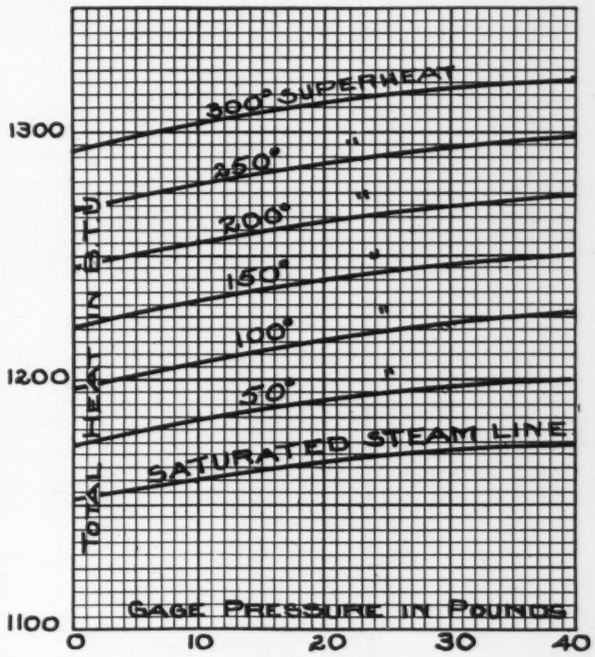


FIG. 5—TOTAL HEAT VALUES FOR DIFFERENT DEGREES OF SUPERHEAT AT LOW PRESSURE

high pressure steam. When high pressure steam passes through a pressure reducing valve and the pressure drops, the difference in the total heats at the two pressures is available for vaporizing the water existing in the steam. Then, after vaporization is complete, the remaining heat will superheat the steam in the pipe or chamber.

Steam traveling through mains at very high velocities likewise suffers a loss of pressure, and this loss of pressure makes available excess heat for vaporizing existing moisture and superheating.

TABLE 3—SPECIFIC HEAT OF SUPERHEATED STEAM

GAGE PRESSURE	SUPERHEAT 5 F	SUPERHEAT 50 F	SUPERHEAT 100 F
1	.471	.468	.466
5	.475	.470	.466
10	.480	.475	.470
25	.497	.490	.483
50	.524	.512	.500
100	.580	.546	.515

Due to the specific heat of superheated steam being somewhat less than one-half that of water, the heat set free in the system from steam in a superheated state is relatively low. Likewise heat made available for superheating will raise the temperature of the steam twice as fast as it would an equal weight of water.

Formulas based on experimental results might be given for computation of specific heats of superheated steam. Since a rather narrow range of values is encountered in heating work, approximate values computed from these formulas are given instead of the complex formula.

Measurements of Evaporation

When evaporation tests or capacity tests are made on heating boilers, it is necessary to reduce all variables to some standard basis. As in all comparison testing, some measuring unit must be adopted.

The one value that remains a fixed quantity is the amount of heat required to change a pound of water at 212 F to a pound of steam at 212 F. This value, as determined by the most exhaustive research, has been found to be 970.2* B.tu.

The *boiler horsepower* is defined as the amount of heat required to evaporate 34½ lb. of water from water at 212 F to steam at 212 F. In actual heat units, this is equal to $34.5 \times 970.2 = 33{,}472$ B.tu.

In boiler operation or testing, the pressures are most commonly other than atmospheric and the return water is not at just 212 F. Furthermore, the steam may contain some moisture or it may even be superheated. Thus conditions might exist where less than 970.2 B.tu are added to each pound evaporated, or conditions might exist where the actual heat added per pound is considerably above 970.2 B.tu.

Under any operating conditions it must be referred to 970.2 B.tu as a standard basis. 970.2 B.tu may be taken then as the value that would give a *factor of evaporation of unity.*

Since actual conditions might give an actual added heat of greater or less than 970.2 B.tu, the number that can be used to multiply 970.2 and give the standard or equivalent evaporation is called the *factor of evaporation.*

Every change of return water temperature or of boiler pressure or of steam quality gives a different factor of evaporation. The following problem explains how this factor of evaporation and the boiler horsepower can be computed when the steam quality is assumed to be 100 per cent.

Problem 1: A heating boiler is operating on return water at 190 F and a steam pressure of 5 lb. gage at

* Goodenough's steam tables give this figure as 971.7 B.t.u.

100 per cent quality. If the boiler evaporates 300 lb. per hour, how many boiler horsepower will be developed?

Computation: From Table 1, the temperature of the boiling point at 5 lb. pressure is 227 F.

To bring each pound of water up to 227 F from 190 F will require 37 B.tu, since the specific heat of water is approximately 1. From Table 2, the latent heat of vaporization at 5 lb. gage pressure is 961 B.tu.

The total heat that must be added to raise one pound of water from the condition of water at 190 F to that

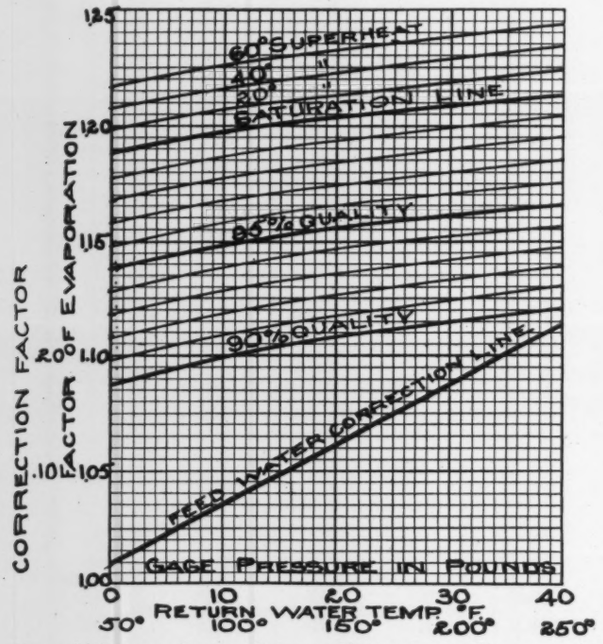


Fig. 6—Inner Scale Shows Factors of Evaporation Plotted Against Gage Pressure. Outer Scale Shows Correction to Be Subtracted for Return Water Temperature Above 32 F

of steam at 227 F will be then the sum of 37 B.tu and 961 B.tu or 998 B.tu.

For a factor of evaporation of 1, it requires 970.2 B.tu.

For the conditions of the problem the factor of evaporation will be $998 \div 970.2 = 1.029$.

Since 300 pounds per hour were evaporated under these conditions, this is equivalent to 300×1.029 or 308.7 lb. under standard conditions.

Since 34½ lb. of water per hour must be evaporated under standard conditions to equal one boiler horsepower, the above boiler is producing $308.7 \div 34\frac{1}{2}$ or 8.95 boiler horsepower.

Correcting for Quality

When the quality of steam is less than 100 per cent, the correction can be obtained by multiplying the latent heat of vaporization by the per cent of steam and adding this quantity to the heat added to the liquid instead of the latent heat of vaporization at 100 per cent quality.

Problem 2: What would have been the boiler horsepower in Problem 1 if the quality had been 95 per cent and all other values had remained the same?

Computation: The amount added to the liquid to bring it from 190 F to 227 F would remain the same or 37 B.tu.

The heat added for vaporization will only be 95 per cent as much, since only 95 per cent is vaporized. The latent heat of vaporization will be therefore $.95 \times 961$ or 913 B.tu.

The total heat required will be 37 B.tu + 913 B.tu or 950 B.tu.

The factor of evaporation will equal, therefore, $950 \div 970.2 = .979$.

The equivalent evaporation will equal $.979 \times 300 = 293.7$ lb.

The boiler horsepower developed will be $293.7 \div 34.5 = 8.51$.

Value of the Factor of Evaporation Chart

Charted values simplify the work of determining the *factor of evaporation* under different conditions. Unfortunately, most of such charts are not adaptable to low pressure heating, since their range of temperature is beyond that of the low pressure heating system. The chart of Fig. 6 gives a range of values commonly found in low pressure practice.

The values given on the ordinates for the different pressures read on the abscissa are for return water temperature of 32 F. This is the temperature from which most steam tables are originated.

But in heating work, most return water temperatures are much higher than 32 F. The values plotted on the straight line on the lower end of the chart are corrections to be subtracted from the chart readings for different return water temperatures.

Problem 3: Determine from chart Fig. 6 the factor of evaporation for data in Problem 1.

Computation: From chart Fig. 6, the factor of evaporation for 5 lb. gage pressure and 100 per cent quality with 32 F return water is 1.193.

The correction factor to be subtracted as read from chart for 190 F is .164.

The factor of evaporation will be then $1.193 - .164 = 1.029$.

Problem 4: Determine from Fig. 6 the factor of evaporation for Problem 2.

Computation: From Fig. 6 the factor of evaporation for steam of 95 per cent quality and 5 lb. gage pressure is 1.143.

The correction to be subtracted for feed water temperature of 190 F as given on the chart is .164.

The factor of evaporation will equal then $1.143 - .164 = .979$.

Flow of Steam Through Pipes

Steam flow in pipes has been a subject of investigation for decades. A host of formulas are available to enable the designer to compute the weight of steam flowing through a pipe in a given length of time. While many of these are reasonably accurate, yet all involve so many variables in the equation that it is necessary to introduce a correction constant to adapt the formula to different conditions, and also to function as a factor of ignorance.

Steam flows from one point to another point of lower pressure. It must be said, then, that drop in pressure causes steam to flow through a pipe.

The pressure gradually diminishes from inlet to outlet

due to friction, deflections, current eddies and changes in pipe sizes. A relatively small part of the pressure drop is required, also, to give the steam its initial velocity.

Since pressure drop is the cause of steam flow, the amount of drop in pressure in a certain length of pipe must be determined if the discharge is to be computed. For high pressure steam it is common practice to allow a pressure drop of one pound for each two hundred forty diameters of pipe. In other words, a drop of one pound is allowed for a piece of pipe equal in length to 240 times the diameter of the pipe.

For low pressure work, where there is not as great a pressure available, this allowance would be too great. This has led to the general allowance of a drop of one lb. in each 1,000 ft. of pipe length for low pressure and heating work.

The general equation applied is:

$$W = C \sqrt{\frac{(P_1 - P_2) - \frac{1}{V} d^5}{L}}.$$

W = Weight of steam flowing per minute.

C = A constant depending upon the size and kind of pipe.

$(P_1 - P_2)$ = Drop in pressure in pounds.

d = Diameter of pipe in inches.

V = Mean volume of one pound of steam.

L = Length of pipe in feet.

For a drop of 1 lb. in one thousand feet of length this formula becomes

$$W = C \sqrt{\frac{d^5}{1000\,V}}$$

For either equation, C has the same values. The plotted curve of Fig. 7 gives values of C for different pipe diameters up to ten inches.

The curves of Figs. 8 and 9 give the determined values of steam flow based on this formula with a pressure drop of 1 lb. in 1,000 ft. of length. For example, a 9-in. pipe will carry 6,400 lb. of steam per hour with a pressure drop of 1 lb. in 1,000 ft. when the initial gage pressure is 5 lb.

Correction Factors for Lengths Other Than 1,000 Feet

For lengths other than 1,000 ft., but with a pressure drop of 1 lb., the steam flow in the different sizes of pipes at different pressures plotted will be equal to the indicated amounts multiplied by the correction factor of Table 4.

For example, if the 9-in. pipe mentioned above had a drop of 1 lb. in 600 ft., the steam flowing in the pipe would be 6400 × 1.288 or 8,243 lb.

Correction Factor for Pressure Drop Other Than 1 Lb. in 1,000 Ft. of Pipe

If the length is 1,000 ft. but the drop in the 1,000 ft. is other than 1 lb., then Table 5 will give the correction

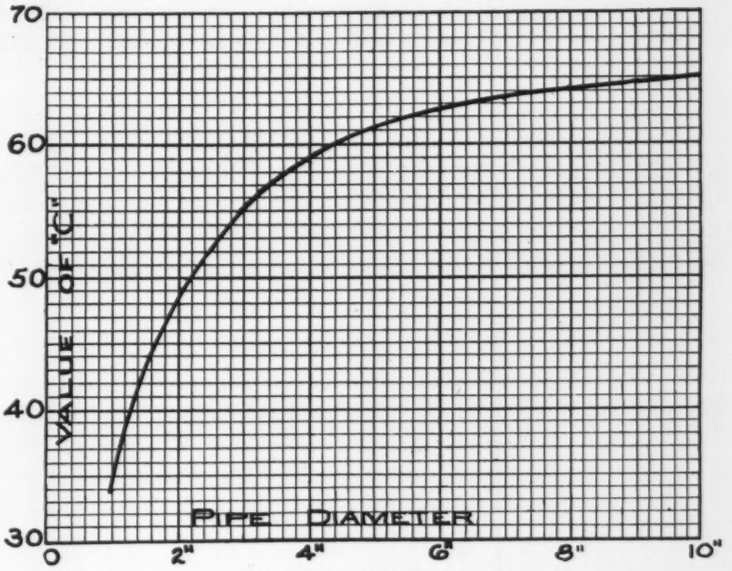


Fig. 7—Values of Correction Factor "C" for Different Pipe Diameters

factor by which the plotted values can be multiplied to give the desired result. For example, if the steam in the 1,000 ft. of 9-in. pipe previously mentioned had dropped 2 lb. in pressure, the actual flow would have equaled 1.414 × 6400 or 9,050 lb. per hour.

In all of the above computations, no allowance is made for condensation of steam due to radiation of heat from the pipes. In case of a long pipe, the amount of steam required to furnish the heat radiated from the pipe must be deducted from the calculated amount of flow or a serious error will be made.

The resistance of valves and pipe fittings must also be taken into account. This resistance may be determined experimentally. If the number of feet of straight pipe which has the same resistance as the fitting is known, then that length can be added to the actual length of the pipe. The sum will be the equivalent

Table 4—Correction Factor for Curves of Flow of Steam for Length Other Than 1,000 Feet

Length in Feet...	50	100	150	200	300	400
Factor.....	4.472	3.162	2.581	2.236	1.825	1.581
Length in Feet...	500	600	800	1000	1200	1500
Factor.....	1.414	1.288	1.118	1.000	0.913	0.817

Table 5—Correction Factor for Curves of Flow of Steam for Pressure Drop Other Than One Pound

Pressure Drop per 1000 ft.	1 oz.	2 oz.	4 oz.	6 oz.	8 oz.	12 oz.	1/10 lb.
Factor.....	0.250	0.354	0.500	0.612	0.707	0.866	0.316
Pressure Drop per 1000 ft.	2/10 lb.	3/10 lb.	5/10 lb.	6/10 lb.	/10 lb.	8/10 lb.	1 lb.
Factor.....	0.447	0.547	0.707	0.773	0.837	0.894	1.000
Pressure Drop per 1000 ft.	1¼ lb.	1½ lb.	1¾ lb.	2 lb.	2½ lb.	3 lb.	4 lb.
Factor.....	1.118	1.225	1.323	1.414	1.581	1.732	2.000
Pressure Drop per 1000 ft.	5 lb.	6 lb.	8 lb.	10 lb.	15 lb.	20 lb.	25 lb.
Factor.....	2.236	2.450	2.828	3.162	3.873	4.472	5.000

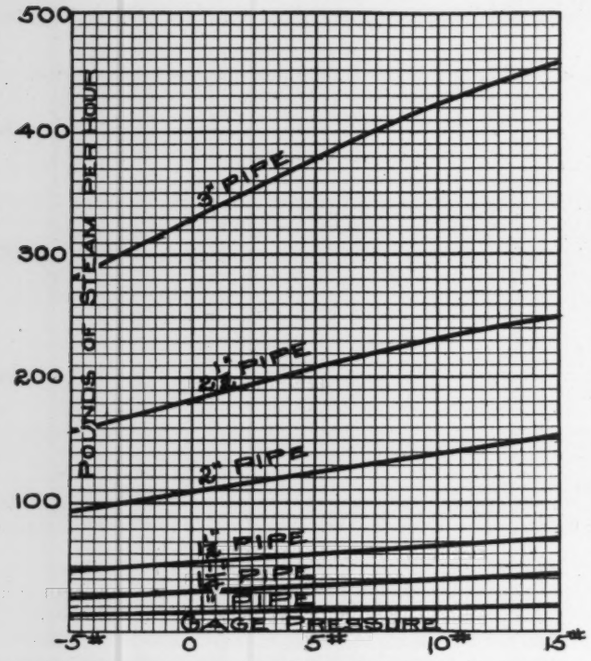


FIG. 8—FLOW OF STEAM PER HOUR THROUGH VARIOUS SIZES OF SMALL PIPE ALLOWING PRESSURE DROP OF 1 LB. PER 1,000 FT. OF PIPE

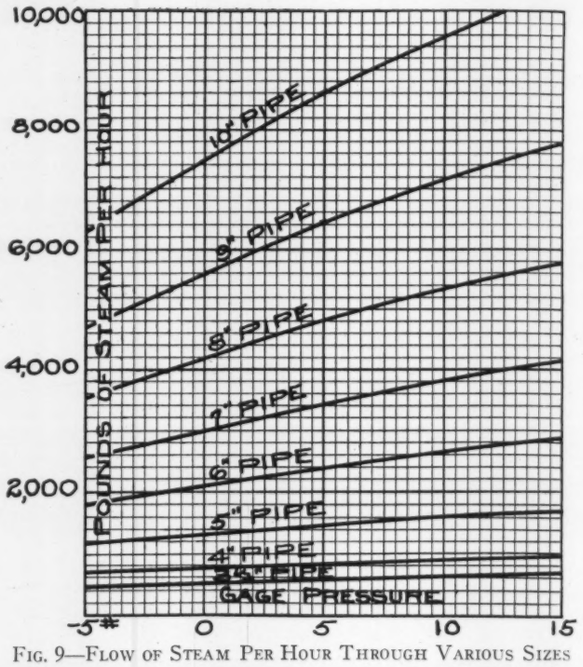


FIG. 9—FLOW OF STEAM PER HOUR THROUGH VARIOUS SIZES OF PIPE ALLOWING A PRESSURE DROP OF 1 LB. PER 1,000 FT. OF PIPE

length and must be used in calculations. Table 6 gives the resistance of fittings expressed in feet of straight pipe to be added to the actual length.

Problem 5: How much steam will flow per hour through a 3-in. pipe 420 ft. long? The pipe has a covering that reduces the heat loss to 75 B.tu per hour per lineal foot. The line has three standard elbows, two gate valves, and one standard tee. The initial gage pressure is 5 lb., and the drop in pressure is 6 oz.

TABLE 6—RESISTANCE OF FITTINGS EXPRESSED IN FEET OF STRAIGHT PIPE

Size of Pipe	Gate Valve	Long Radius Elbow or Run of Standard Tee	Medium Radius Elbow or Run of Reduced Tee	Standard Elbow or Run of Much Reduced Tee	Angle Valve	Short Return Bend	Side Outlet Tee	Globe Valve
2	1.7	2.3	3.2	4.3	7.1	8.2	12.1	14.8
2½	2.4	3.1	3.9	5.7	9.5	11.1	15.1	20.0
3	2.6	3.9	4.8	7.8	12.0	14.0	19.2	25.0
3½	3.3	4.7	6.2	9.4	14.1	15.1	22.8	30.0
4	4.0	5.5	7.0	11.0	16.4	18.0	26.0	35.0
5	5.5	7.1	9.8	14.2	21.5	22.8	22.3	45.0
6	6.9	8.4	11.1	17.5	24.7	27.8	38.3	54.0
7	7.9	10.3	13.4	20.7	28.9	32.0	43.3	64.0
8	9.4	12.0	15.6	24.5	33.1	37.0	49.5	74.0
9	10.3	13.9	17.5	27.8	37.5	31.5	55.4	84.0
10	12.0	15.8	19.4	31.2	42.0	46.0	61.3	94.0

Computation: Resistance of fittings (From Table 6):

3 Standard elbows	$7.8 \times 3 = 23.4$
2 Gate valves	$2.6 \times 2 = 5.2$
1 Standard tee	$3.9 \times 1 = 3.9$
	32.5

Equivalent length of straight pipe $= 420 + 32.5 = 452.5$

From Fig. 8, the flow would be 379 lb. for 1 lb. drop in pressure per 1,000 ft.

From Table 5, the correction factor for 6 oz. drop in pressure is 0.612.

From Table 4, the correction factor for 452.5 feet length is 1.49 by interpolation.

Without loss, then, the flow will be

$$379 \times 0.612 \times 1.49 = 345.6 \text{ lb.}$$

The heat loss will be $75 \times 420 = 31{,}500$ B.tu.

From Table 2 the heat of vaporizaton of one pound of steam at five pounds pressure is 961 B.tu.

The pounds of steam condensed will be then

$$31{,}500 \div 961 = 32.7$$

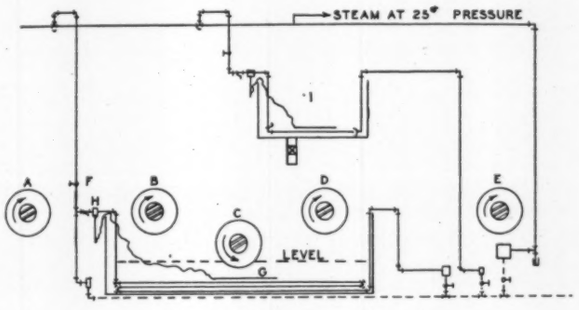
If 345.6 lb. of steam started through the pipe and the condensation loss was 32.7 lb., the amount of steam delivered will be $345.6 - 32.7 = 312.9$ lb.

Temperature Control on Paraffine Paper Machines

THE superintendent of an eastern concern specializing in the manufacture of bread, butter and creamery packing paper, came to the conclusion a few months ago that he could secure a considerable saving in paraffine consumption if he could keep the paraffine bath in the paraffine machines at a certain temperature.

After a few weeks of intensive research work and testing this temperature was determined to be 140 deg. F.

A sketch of a paraffine paper machine is shown herewith. It consists mainly of a steel tank with a pipe coil covering its bottom. The paper is brought in a large roll and mounted on drum *A*. It then travels under a fairly good tension over the side of drum *B*, the lower side of *C* and thus through the paraffine, and finally, after going over drum *D*, is rolled up again on *E*. All the drums are driven by one motor.



A Paraffine Paper Machine

Up to the time the survey was made, the gate valve *F* was manually operated by the attendant to keep the temperature of the paraffine fairly constant, not with thermometers but simply by watching the color of the paraffine. None of the lines were trapped and the feed to each machine was taken out of the bottom of the overhead supply steam main.

To make a good temperature control installation for these machines proved to be harder than anticipated. The difficulty arose from the irregular running of the machine. It takes about fifteen to twenty minutes for one roll of paper to pass through at maximum speed. Considerable heat was taken up during each run; first by the paper itself and second by the cold cubes of paraffine which were placed into the bath at intervals to keep the bath at a pre-determined level. After a roll had run through, it took about five minutes to take it off drum *E* and mount a fresh roll on drum *A*. Due to the large coil which was necessary for a regular run under the above conditions, the temperature of the bath went way up during the idle minutes and even a fairly tight valve *F* would not prevent this temperature rise.

The job to obtain a constant temperature was tackled by first changing the piping to that shown in the sketch; steam supply to each machine was taken off the top of the main and the latter, as well as the first, was drained into a return line through thermostatic traps. A self-contained temperature regulator was then installed with the thermostat at *G* and the piston valve at *H*, both connected by a flexible tubing. Although this arrangement kept the temperature of the paraffine practically constant when the operation was going on, it was found that the steam already in the coil of the tank heated the paraffine well above the desired point at the time the machine was shut down for renewing the paper roll, even though the regulator valve *H* was shut off.

This was due, naturally, to the large heating surface of the steam coil. However, a smaller pipe coil would not do, as it could not possibly supply enough heat to melt the fresh cubes of paraffine while the machine was in operation.

The final solution of the problem was the addition of a paraffine melting tank *I*. In this tank the cold paraffine was melted and brought up to the operating temperature and held there by means of another temperature regulator. Paraffine was then supplied to the machine as needed and at the right temperature. The heating coil in the machine could thus be reduced considerably and the regulator was able to hold a constant temperature even when the machine was standing idle.

This installation secured the company an A-1 product, cut down their paraffine consumption, and, while heretofore one operator was needed for every two machines, he could now take care of five.—*E. M. Mittendorf.*

Salt Drying

THE removal of moisture from salt crystals, following their production from liquor in the evaporator, presents an interesting problem. Two methods have been used in the past. One might be classified as mechanical separation; the other is a vaporization process.

It might be interesting and instructive to examine the steps necessary in the removal of this moisture, from a theoretical viewpoint.

In the first place, this moisture on the surface of the crystals has a vapor pressure somewhat lower than that of water. The salt crystals themselves have a vapor pressure which is almost negligible.

Suppose, for the sake of argument, the salt is of the common table variety; then the moisture consists of water and dissolved sodium chloride. The water will occupy half of the exposed surface of the moisture and has, therefore, a 50 per cent chance to escape. Equilibrium with the water vapor in the surrounding air will thus be had at a vapor pressure, for the water in the moisture, of one-half that for water at the temperature which prevails. In other words at a relative humidity of 50 per cent, which is exactly the same thing as a relative vapor pressure of the same amount, equilibrium is established.

In order that vaporization can take place, the relative humidity of the air must be lower than 50 per cent and the greater the difference between 50 per cent and the prevailing relative humidity the greater the rate of drying. It is therefore evident that such salt crystals could be dried very rapidly at very low humidities.

In order to establish the humidity which prevails around salt crystals which are drying it is necessary to know the temperature at this point. The next step is to measure the relative humidity and temperature of the entering and the leaving air. It will then be easy to calculate the relative humidity which prevails around the crystals. It will then be possible to determine just how much higher the temperature can be raised in order that the relative humidity in the drier can be lowered and the rate of drying increased.

Drying salt crystals by means of air conditioning, using refrigeration, might not be able to compete with the furnace-rotary drier method. Nevertheless it is possible that air conditioning which employs gels for the absorption of water would furnish a method of drying which could be controlled with great ease and which would eliminate the danger of contamination with products of combustion.—*M. T.*

Testing Circulation of Air

FIG. 1—THE GENERATOR WITH THE LEAD HEADER REMOVED TO SHOW ITS CONSTRUCTION. THE MOUTHS OF THE GLASS JARS ARE CLOSED WITH LARGE CORKS. BOILING THE CORKS IN PARAFFINE PROTECTS THEM FROM THE CORROSIVE ACTION OF THE CHEMICALS

WHAT actually takes place in ventilating systems is frequently very difficult to determine by observation or by air current measurements. Anemometer readings at various intake and outlet openings are of value in determining gross air movement, but such readings give little indication of the circulation of air or the path of air movement through the space of the room in question. Backdrafts, eddy currents and air pockets must be located by some other means.

In tracing air movements through cold storage plants, farm fruit storage and other buildings, the writers have found the rough apparatus described below of great value. This apparatus is a simple device that can be made at a nominal cost and takes advantage of the well known chemical reaction in which the vapor of ammonium hydroxide is passed over concentrated hydrochloric acid causing the white, billowy fumes or fog of ammonium chloride to be blown into the air, according to the following reaction:

$NH_3 + HCl = NH_4Cl$ (ammonium chloride)

The ammonium chloride thus formed is a very finely divided precipitate or powder, so light that it diffuses readily, riding upon the air currents and indicating the direction of air movements in a very positive manner. The "smoke" or "fog" produced has no appreciable heat of its own to cause it to form convection currents as may be the case when warm smoke is used. It is readily discernible in the air in very small amounts and its path of diffusion through the air of a room may easily be determined.

When used even in relatively much larger amounts than is necessary to make the required observations, there appears to be no objectional residue in the form of dust or any marked discomfort in breathing the atmosphere. During a series of poultry house ventilation studies, chickens were kept in houses in which the air was "fogged" in this manner continually for five days and nights without injury to the fowls or any indications of discomfort.

Assembly of Apparatus

The apparatus as assembled by the writers is composed of two containers for the chemicals consisting of large-mouthed, glass candy or chewing gum jars, five inches square and about six inches high, mounted in one end of a box-like wooden frame with a tight fitting compartment for each jar. Battery jars might be used if desired. Connecting the two jars is a lead pipe header with an entrance pipe to the first jar, a pipe connecting the first and second jar and an outlet from the second jar. These pipes are held in lead covers provided with a flange so that the whole may be fit snugly into the jars when the apparatus is in use and removed when the machine is idle. The openings of the jars are closed with large, tight-fitting corks to protect the contents and to confine the vapors of the ammonia and the hydrochloric acid, either of which alone is unpleasant. The other portion of the apparatus consists of a small bellows mounted on the rear portion of the box in such a way that the mouth of the bellows fits into the inlet opening of the lead pipe connector when the latter is in place. The portion of the box beneath the bellows is of a convenient length depending upon the size of the bellows and is left partially open at the end, serving as a general utility compartment for the corks, gloves or other accessories. Handles made from number nine galvanized wire are attached to the box as shown in the illustrations so that they fold back out of the way when the machine is in operation. Care must be taken, of course, not to splash or spill either of the chemicals upon the clothes or skin, as both solutions are very caustic and likely to cause injury. Reserve supplies of each solution should be kept in a safe place in tightly stoppered bottles properly labeled. Strong ammonium hydroxide should be secured, as many of the weaker "ammonia waters" sold as cleaning solutions are not satisfactory for this purpose.

Operation

A gentle manipulation of the bellows forces a stream of air into the first jar, which contains the concentrated ammonium hydroxide, and from this jar to the next, which holds the concentrated hydrochloric acid. A dense fog ensues from the outlet tube, the volume and density of which may be controlled by the operator through the manipulation of the bellows. For most operating conditions a series of quick, short strokes is most satisfactory and will cause the emission of a uniform fog which will diffuse gently from the mouth of the apparatus. This machine has a large capacity and it is possible to fill a large room in a very short time. An advantage secured from careful and gentle operation of the bellows is that the natural air currents of the room under observation

By I. D. Mayer*
and C. E. Baker*

The chemical fog generator described in this article has many uses for consulting and maintenance engineers and contractors. Frequently questions arise regarding the direction, speed and distribution of air movement in ventilating systems that are difficult to answer, even by the use of elaborate instruments. This simple device may be used:

(1) To determine the efficiency of ventilating systems by actually tracing the movements of air currents through the space being ventilated.

(2) To determine the actual frequency of complete air changes in any type of forced air or natural ventilation system.

(3) To locate "pockets" or areas of low or inactive air movement in ventilated rooms.

(4) To locate undesirable "drafts" and direct air movements in auditoriums, schools and other gathering places.

(5) To test the tightness of ducts and air passages of forced air circulation systems or to locate air leaks in large rooms or compartments.

(6) To locate convection currents set up by different temperature gradients in different sections of a given space. As the fog produced has no appreciable heat, and may be introduced with little disturbance of the natural air movement, its use along this line presents many possibilities.

Many other specific uses might be enumerated.

are not disturbed. The capacity of this apparatus may be further increased by using an electric fan or blower to force the air through the jars, but such a practice introduces a factor of an unnatural air current which may cause an error in some types of observations.

Introducing the Fog

The chemical fog may be introduced at several points in the ventilating system. In plants with natural air movement, such as ordinary cold storage plants in which the convection currents set up between the cold coils and the warmer room are depended upon to furnish circulation, the fog may be simply blown into the room and its movement observed. The rate of motion can readily be measured under most conditions even when the room is filled with the fog and the air pockets or other irregularities quickly located. If some doubt exists as to the air movement conditions of a certain part of the room, the fog may be evacuated or permitted to settle and another smoke test made in the portion of the room in question, noting the direction of air movement and the rate of diffusion. Under these conditions the true air movements may readily be determined.

In plants using forced air circulation, the fog may be introduced into the air ducts or openings before they enter the room under observation. The entry of the fog into the room may be noticed and its movement followed with great accuracy. In systems using air washing equipment or dehumidifiers the fog should be injected into the system after the air has passed these devices.

Finding Frequency of Air Changes

In determining the frequency of air changes in rooms or buildings this system also may be used to great advantage. Calculations based upon volume air movements do not accurately represent actual air changes. They merely mean that a volume of air equal to the cubical capacity of the room passes through the intake or outlet system every so often. The true air change or time of complete replacement may be a very different matter. By filling the room with a uniform fog and then observing the length of time required to clear the air with the ventilating system in operation, a much more accurate idea of the frequency of air changes may be determined. In an efficiently designed ventilating system the frequency of air changes as determined by the displacement method may check very closely with the calculated frequency. If the system is not efficient there may be a wide variation.

Detecting Leakage

The leakage from air ducts or passages as well as from compartments of various sizes may also be determined by similar means in many instances. Leaks from rooms may be detected by filling the room with a dense fog which is kept in circulation within the room. In a tight room, the fog should remain indefinitely, but if leaks are present the fog will gradually disappear, the rapidity of its disappearance depending upon the severity of the leaks. As the finely divided ammonium chloride tends to settle out of still air, an active circulation or agitation must be maintained within the room. Even with such a circulation the air will eventually clear from the settling out of the particles, so that this fact must be given due consideration.

Heat leaks may be studied in a similar manner, but where the loss of fog may be observed from a point outside of the room or space being observed a more accurate estimate of conditions may be made and the definite locations of leaks determined.

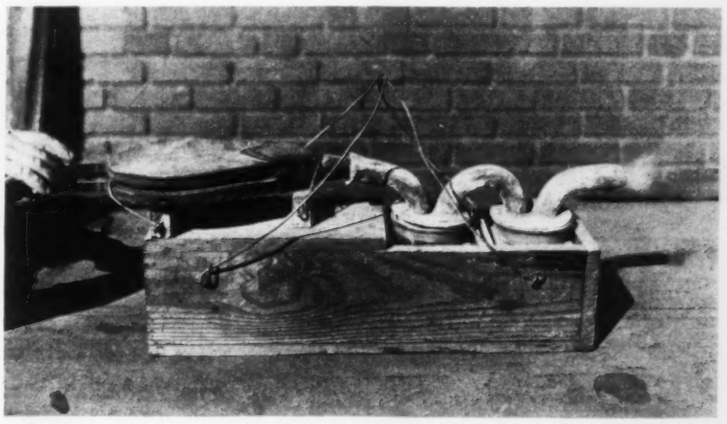


FIG. 2—THE "FOG" GENERATOR READY FOR ACTION. A GENTLE MANIPULATION OF THE BELLOWS FORCES A CLOUD OF WHITE FOG FROM THE OUTLET

* Purdue University Agricultural Experiment Station.

Specifications for Heating and Ventilating

By Samuel R. Lewis

Article No. 2

THERE are a few legal requirements which should be covered in preparing specifications, but I do not hold to the school which attempts to describe every screw. My experience has been that when the contractor and the engineer have the desirable mutual spirit of helpfulness they will turn out good results without much recourse to a fine-tooth comb worked through the specifications. When the engineer and the contractor lack the spirit of helpfulness, no specification can be so smooth and perfect as to emerge unscathed from the charged atmosphere of dissension. Human frailty is such that usually when we find an engineer who writes a copper plate specification, developed in minute detail, we also find the drawings which accompany the specification lacking in study and skill, especially on the larger items of general arrangement and selection of materials.

We often see much of the matter which might be placed on the drawings described in the specifications, and some engineers cover the drawings with notes and sketches so as to achieve the least possible in the specifications.

It may be that the drawings should show the volumes of material and the proposed location of it, while the specifications should describe the quality of material and workmanship.

In the following article I have recorded some of the outstanding points under each item which experience suggests need to be cleared in the specifications by the engineer. This is the second part of the article on specifications; the first appeared in the *March, 1930,* issue of HEATING, PIPING AND AIR CONDITIONING.

Test of Piping

Piping must be tested before it may be covered up, and before insulation can be applied. The test pressure should exceed the maximum operating pressure.

Insulation Inside Buildings

This refers especially to insulation which will prevent unnecessary waste of fuel, which will prevent overheating, and which will reduce fire hazard.

The boilers and breeching, almost without exception, should be insulated heavily, the former to conserve fuel, the latter to keep the boiler room cool and to increase stack temperature and draft. The insulation contractor usually furnishes such steel and wire and drilling and fitting as is necessary to attach the insulation permanently. It is wise, on breeching insulation especially, to provide in the insulation for expansion and contraction joints. Breeching insulation should not have canvas outside of it but may be finished with hard cement or with wire screening, since canvas can burn if ignited.

Boiler insulation should be protected with heavy metal where traffic or drippings or floor-flushing might wear it or dampen it. It is good practice to jacket all boiler-insulation entirely with metal.

Pipe insulation is often sectional and may be applied with pasted canvas and with lacquered metal bands. An installation may be made with bands only a few inches apart, of copper wire, with canvas sewed on.

In pipe shafts which communicate through from floor to floor of lofty buildings, no canvas jackets should be used on pipe insulation, since this canvas creates a very real fire hazard and has caused fires. Wire screening or hard plaster finish may be used instead of canvas.

Cold pipes and tanks must be insulated, both against undesirable heat transfer and against condensation drip.

Insulation Outside Buildings

Pipes run outside preferably should be buried underground. Where they must run overhead they should be insulated as recommended by the manufacturers, with usually double or triple the thickness used indoors. The insulation should extend entirely through outside walls, should have all joints alternating or broken, and, of course, must have a waterproof and windproof jacket. Supports should be attached rigidly to the pipes, and may be welded, with allowance for movement entirely outside of the insulation, as by swinging hangers, rollers on tracks, etc.

Where pipes run underground, the same care should be taken to preserve the insulation at supports. Drainage of the conduit is of the utmost importance. No underground conduit for heating pipes ever should be installed without a crushed rock or cinder filter under it with an independent drain pipe. All underground conduit for heating pipes must be waterproof beyond peradventure. There should be no air circulation of any consequence within the conduit. Expansion elements and anchors must all be developed to an exact degree.

Building Insulation

The engineer should be familiar with the benefits to be obtained by promoting better heat and moisture resistance in building construction. Often, in case of trouble, it is wiser to correct the faulty walls or windows or ceilings than to install additional heating capacity.

Fans

The specifications should state the output desired, at the pressure-resistance expected, and should give the power required. They may state the make and the size and type and speed of the fans. They may go further and name the weights and the gages of metal to be used

in the fan construction. When housed fans are called for, the extent of the housing, whether full or seven-eights or three-quarters, and the point of discharge should be stated, especially if this is not shown clearly on the drawings. It makes a difference in cost whether the wheel or the pulley are overhung. It is always desirable, when practicable, to have the bearings out of the air-ways. The smaller sizes of fans are now being fitted with ball and roller bearings; these are used on many installations. It is practicable to provide throttling dampers on the outlets of housed fans, since the power consumption is proportional to the amount of air delivered; and temporary reductions in output thus can be made with corresponding power savings.

Power Transmission

The endless belt is a reliable and low cost transmission-servant where the direct-connected apparatus cannot be used. There are several proprietary short-center special belts and chains which give excellent results and which are quiet and durable.

Motive Power

The motive power usually is electric. It is often expensive and difficult to secure variable speed motive power for fans and in many cases where definite air volumes must be provided, it is not desirable to use variable speed. Where the desirable speeds for fan and motor do not coincide, belts may be used. An advantage with belts is that the speed ratio easily may be changed at little trouble or cost by changing the pulleys.

Steam or internal combustion engines frequently are used for driving fans. Steam engines are justified economically where there is use for the exhaust steam, and they form a very quiet and flexible motive power for fan-driving.

The specification should state the kind and the speed and the output of the motive power, and should determine the method of speed control.

With electric power, protection against overload, low-voltage and phase-splitting should be called for.

Sound Proofing

The best practice calls for the foundation for the fans and the motive power to be integral and of considerable inherent inertia. This foundation should be entirely separate from the building-structure. It, with its burden, preferably is floated in a bed of some elastic and non-resonant material such as rubber or cork or vegetable fiber. All contacts between the insulated machinery and the ducts or partitions should have sound-inert inserts, such as canvas or leather or rubber.

It is often effective with noisy basement fans and with oil burner equipment to suspend below the normal basement ceiling a second ceiling a few inches lower. The installation of a plastered ceiling below either wood or concrete joists assists remarkably in stopping sound-transmission from machinery in basements.

There are several proprietary systems of sound-proofing for machinery, and these can be relied upon in almost all cases, though very careful supervision of workmanship by all parties concerned is important. Loosely-woven or felted material, whether in quilt or in board or in fill form, is useful for sound-insulation.

Air noises in ducts often can be filtered out by passing the air through felt-faced labyrinths. The degree to which a sound insulator in an air-way absorbs dust, or is fire resistant, is a measure of its value for the purpose.

It has been found effective in sound-insulating small fan rooms on the upper stories of buildings to line the entire room with felt and to protect the felt with a sheet steel facing; the sheet steel, however, not making any contact with the building structure except through the heavy felt.

Convectors

The use of this word is more logical than the term "indirect heaters" or "indirect radiators." The specification should cover methods of support, air venting, encasement, access, and, of course, must describe the subdividing to obtain control. With nonferrous convectors that are light in weight and comparatively small in interior capacity, precautions must be taken to prevent freezing.

One approved method of protection against freezing is to have not less than two sections of tempering heater, the outer one capable of bringing the entering air from the coldest on record to a point above freezing, and the inner one capable of bringing the air from the point

where the outer section leaves off, to the temperature desired. The outer section is controlled by a duct thermostat on the cold side, so arranged that the supply valve shall always be wide open unless the thermostat receives motive power. If the thermostat, receiving motive power, elects to permit that power to pass it, the valve can be closed, but under all other conditions the valve is open.

The inner section is controlled by a conventional duct thermostat usually in the fan outlet. Each different kind of convector presents problems of its own as to drainage and air venting.

It is wise to insure against expensive tearing out in the event of a freeze-up by exercising discretion as to double and triple decking or super-imposing of blast convectors.

Direct Heaters

Direct heaters usually are of cast iron. Do we wish to hang them from the walls or from the window frames, or do we wish extra-high legs?

It is convenient and desirable to have the radiators factory-dipped in a priming coat of paint. The quality of this paint is of importance. It is wise to insist that a very careful check be made by the contractor before ordering the radiators, to be sure that none of them shall project above the window stools.

Do we wish covers or enclosures?

Concealed Heaters

These should, as a rule, be delivered complete with built-on metal encasement in which provision is made for access to valves and traps and to the heater-sections, with flanges for attachment of grilles. It is not satisfactory to place dependence upon unlined flues or recesses for these heaters.

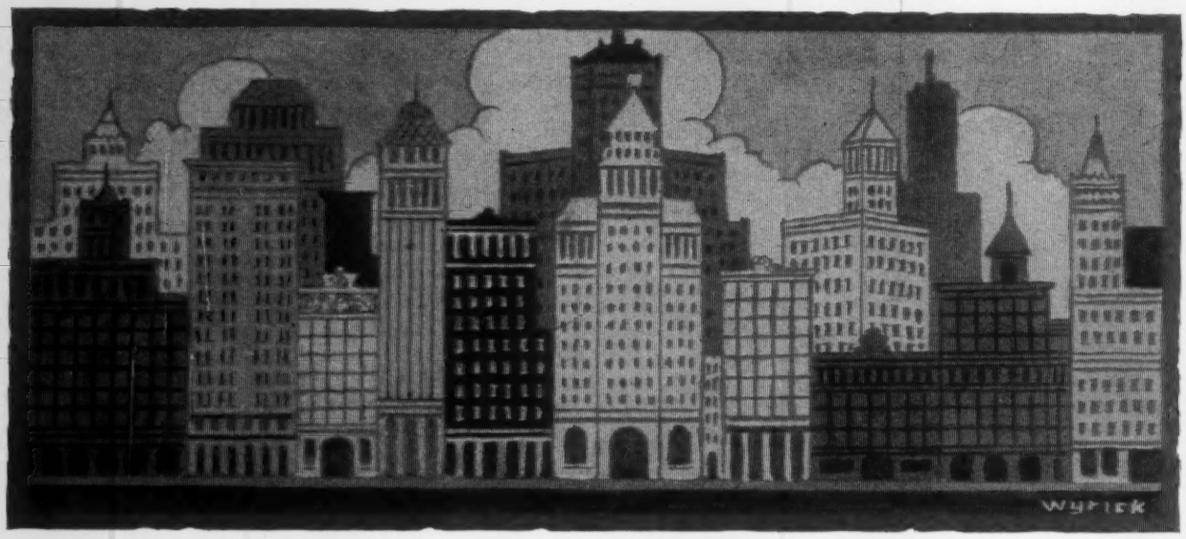
Partitions and Housings

These should be detailed and described carefully. They usually must have frequent structural steel braces all around their edges and not over 24 in. apart vertically.

They must have well braced doors with adequate hardware and must be fitted tightly around pipes and ducts and shafts, etc. Usually all sections in partitions permitting removal of convectors or machinery should be assembled with bolts.

Sheet Metal Duct Work

The kind of metal, the gages, the seams, the bracing and the supports all must be described. There must be no vibration under full capacity and the ducts must be air tight. Flanges should be left to receive all connections, and all registers and grilles. The contractor must coordinate with the piping contractors so as to avoid



interferences. The contractor must install pneumatic dampers, thermometers, etc., furnished by other contractors.

Inlet and Outlet Hoods

Intake openings preferably are to be placed in side walls, where the cleanest air usually is obtainable. Bird screens and protection from rain and snow must be provided. Louvres of non-corrosive metal or an overhanging hood may be used. An intake directly above a dusty roof is objectionable on account of the dirt from wind eddies as well as because the entering air will be unduly heated on many a sunny day.

Outlets are most effective when in the form of chimneys looking straight up. Any hood forms something of an obstruction.

Outlets from ventilating systems must not look out horizontally, even though they have exhaust fans attached, since they become reverse-acting funnels whenever they look to windward. An outlet, even though looking up and provided with the best hood practicable, will cause trouble due to back draft if close to the wall of a taller building and on the windward side of it. It is always wise, when confronted with such situations, to extend the ducts to the top of the more lofty building.

Guards

Guards should be specified for all danger spots, such as fan inlets, belts, pulleys, pits, etc. These are required by law in many localities and always are a proper requirement.

Dampers

There must always be means of closing the main intake and outlet openings for periods when the ventilation

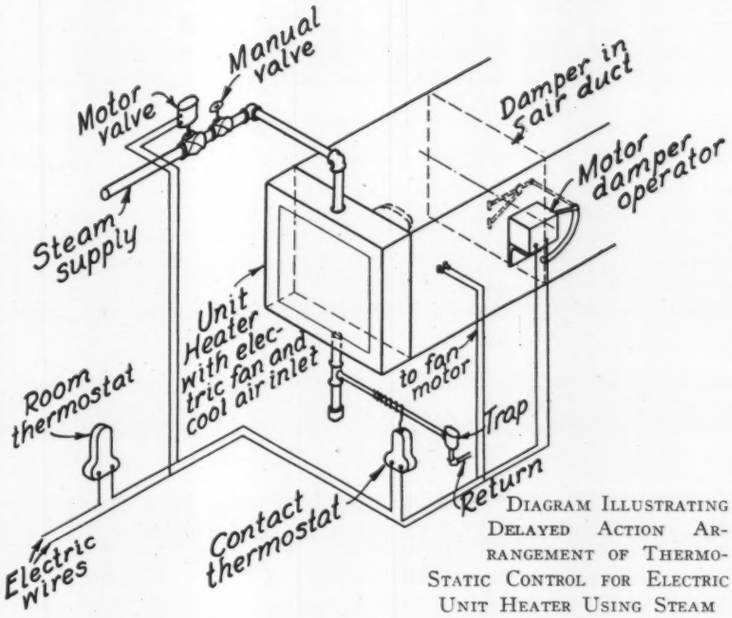


Diagram Illustrating Delayed Action Arrangement of Thermo-Static Control for Electric Unit Heater Using Steam

is shut down. While pneumatically-operated dampers can be used for this purpose they cannot easily be made perfectly tight, may not hold against heavy storms, and are dependent upon springs against compressed air, which may fail.

There is a decided tendency, therefore, to use manually-operated main dampers with steel cables, pulleys and winches where easy access cannot be had.

Volume dampers always are manually controlled. They should be of non-rattling type, with means of locking in place. It is highly desirable that there shall be a hand-hole in the duct adjacent to every damper, since sooner or later it may be necessary to inspect the fastenings or to tighten worn bearings. Volume dampers should always be as remote from the outlet openings to the rooms as is practicable, so that the reduced air flow in the duct may stabilize itself after the disturbance created by the damper.

Registers, Grilles and Diffusers

Registers, grilles and diffusers should be specified as to material and style and the method of fastening them in place should be detailed carefully. Serious injury to people has been caused from carelessly fastened apparatus.

A reform in nomenclature is needed. It is suggested that the following be adopted:

Grille: A covering or finish plate for an air way opening having perforations to permit air passage. It may be of cast or wrought metal or may be of wire or plaster or wood or of any other suitable material. It may be a "register face" but will have no damper or register body attached to it.

Register: A louvred box behind a grille called a register body, adjustable for regulating air flow in combination with a face plate or grille. The word register is not to be taken to mean the grille only, but always means the grille and the register body in complete assembly. The register may be of cast metal or of wrought metal.

Diffuser: A structure which fulfills part of the function of a grille, and which may be attached to the front or back of a grille or which may substitute for a grille. It may be of any suitable material or design.

When humidifying apparatus is used it is wise to require non-corrosive materials in the construction of grilles, registers and diffusers.

There are many proprietary types of inlet and outlet covering made in different metals and designs to supply special needs.

Ash and Coal Handling

The ordinary small heating plant may be served with a simple coal barrow and with ash cans and a barrel truck. It is always desirable to minimize labor and to install mechanical coal and ash handling where the investment can be justified. There are several large organizations which specialize in this equipment, and apparatus is available at reasonable cost which will deliver the coal to the stoker hoppers and which will remove the ashes to a storage bin without any human intervention except the pressing of a button.

Ash hoists which will elevate large cans of ashes either electrically or manually, through hatches looking up or through upper level doors looking out, are available from several manufacturers.

Thermometers and Instruments

A recording pressure gage is an excellent adjunct to most steam heating and ventilating systems, since it checks accurately the quality of the service which the fireman delivers. Thermometers should be provided for indicating the outside temperature and the temperature at each stage of the air conditioning process. These usually are of the extension type. A thermometer in the return water to the vacuum pump usually will disclose whether the traps are functioning satisfactorily, and for this reason such a thermometer should be provided.

Automatic Heat Control Using Compressed Air

It is usual to indicate in the specifications the number and the location of the thermostats and dampers and valves. The compressor generally is electric driven, though steam and hydraulic air compressors are used. The kind of air piping, whether of brass or steel or iron and the kind of thermostat should be determined by the specifications. Diaphragm valves and damper motors may be provided wtih rubber bellows or may have metallic bellows.

Automatic Heat Control Using Electric Apparatus

While complete low-voltage electric automatic temperature regulation is now available, electric equipment is used very widely for regulating automatic fuel burners and for unit heaters which have electric fans. A wide variety of functions becomes possible for this highly specialized control when electricity is employed.

On hot water systems, either for heating buildings or serving baths, etc., an electric aquastat may be used to control an electric circulation-booster. Often, on building-heating hot water systems where there is an oil burner, the booster pump motor is arranged so as to operate only when the oil burner operates.

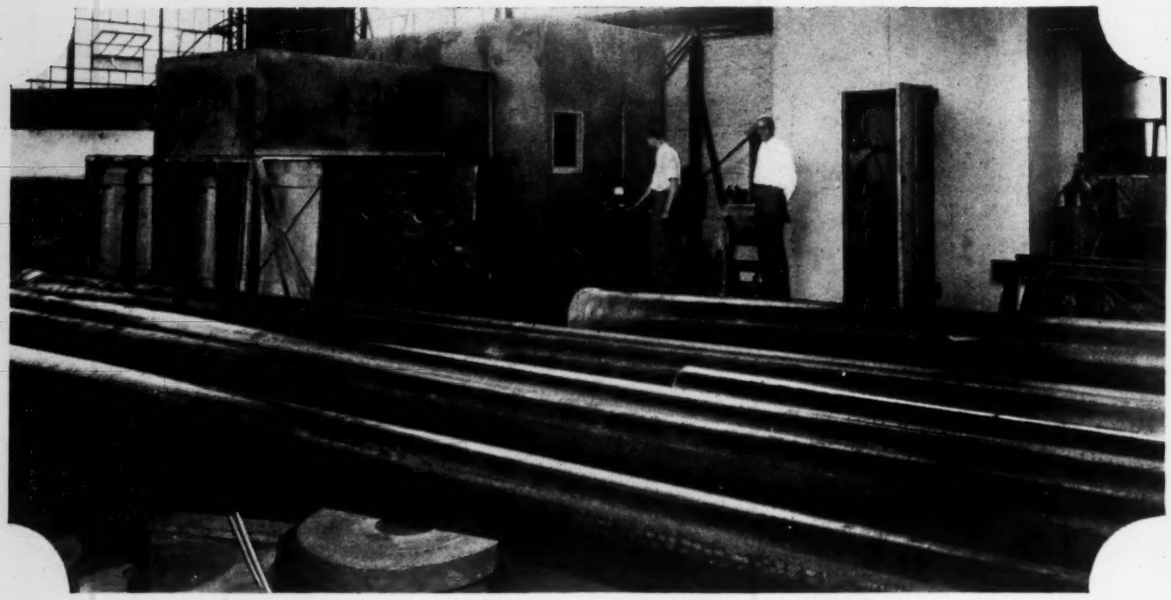


FIG. 1—AS SEVERAL OF THE FITTINGS WEIGHED MORE THAN THREE TONS EACH, IT WAS DECIDED TO CARRY ON THE INSPECTION IN THE PLANT

High Pressure Fittings Inspected with Transportable X-Ray Equipment

By Ancel St. John and Herbert R. Isenburger

DEVELOPMENT in the science of X-raying as a means of scientifically inspecting metals, especially pipe and fittings intended for high pressure service, has been brought to a satisfactory basis. Since the discovery was made thirty-five years ago by Professor Roentgen, this development has kept pace with the forward movement of other efforts similar in scope. This progress has brought about, in this country at least, the plan of applying the X-ray method of examination with what might be termed mobile apparatus set up in the plant of the power piping contractor. For some time several laboratories have existed equipped to carry on this method of flaw detection in metal materials of large dimensions, an outstanding example being that of the United States Government at its Watertown Arsenal, maintained chiefly for the X-ray examination of heavy calibre guns.

Castings Inspected on the Job

In order to be sure of sound steel castings for their 1,200-lb. steam plant, the owners of the station and their engineers decided to have these castings X-ray inspected before putting them into service. As several of the fittings weighed more than 3 tons each, it was decided to carry on the inspection at a place better equipped than the home laboratory for handling such heavy objects. Accordingly, the high potential power plant was transferred to a suitable location near the plant. After completion of the inspection the equipment was returned to the home laboratory. It is believed this is the first time in the history of the art that X-ray equipment, powerful enough to examine steel 3-in. thick, has been used as a transportable unit.

The manifold, intended to carry a working steam

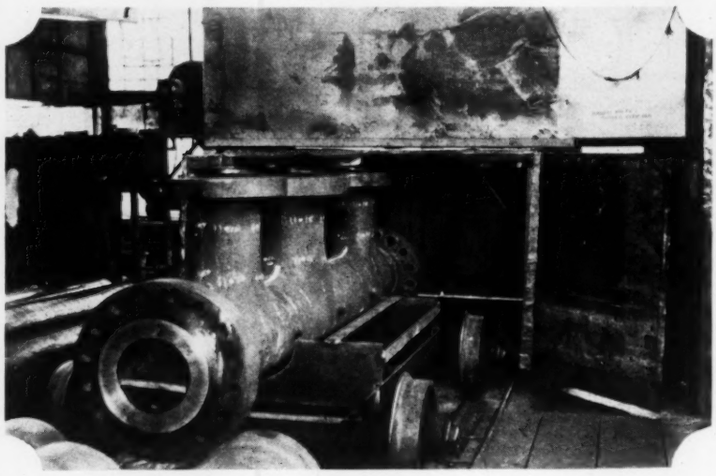


FIG. 2—THE CASTING AND HOUSING TRAVEL ALONG TRACKS

pressure of 1,400 pounds, which is shown in the illustration, is approximately 11 feet in length with a 10½-in. inside diameter, a wall thickness of 2¼ in., from which three nozzles project, each 2 ft. long, with a similar wall thickness. Its weight is about 7,000 pounds.

The Apparatus

The X-ray tube is in a lead-lined box within a sheet iron case directly over the casting under investigation. The photographic films are placed in special cassettes on a carrier which can be seen projecting from the end of the casting. The X-rays thus pass through the wall and make a shadow picture which shows the location and nature of any serious defects. During exposure, the casting is surrounded by a lead-covered housing to prevent X-rays from going where they are not wanted. The casting and housing travel along tracks under the tube; changes in position of the casting are made with a 5-ton crane. The 200,000-volt direct current is produced by special transformers and rectifiers within a sheet iron room just behind the control stand. The high potential equipment is thus completely surrounded by a grounded metal shield. Fig. 2 depicts a large manifold mounted for examination of regions indicated by the white stripes, with the lead housing rolled back out of the way. The entire equipment occupies about 600 sq. ft. of floor space.

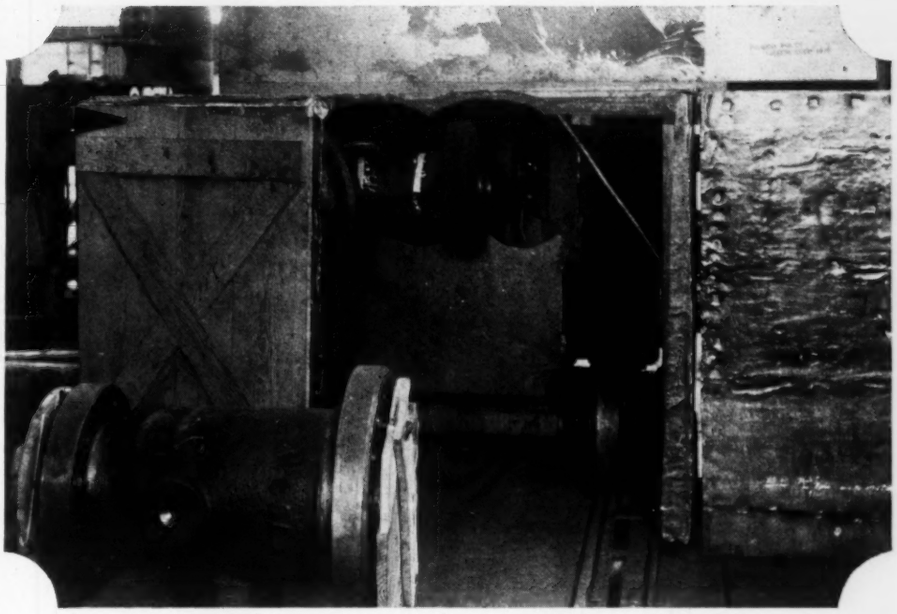


FIG. 3—THE REGIONS TO BE EXAMINED ARE INDICATED BY WHITE STRIPES

Time Required for Inspection

For the complete examination of a fitting of this size, approximately 125 X-ray pictures are required, consuming about thirty hours of actual exposure time and at least ten hours for arranging the casting and placing the films. The development of films and the loading of cassettes are done while exposures are in progress. Smaller fittings require correspondingly shorter times. Other fittings tested included 6-in. tees requiring nineteen pictures, ells, various safety valve bodies and bonnets and return bends, each requiring an average of ten.

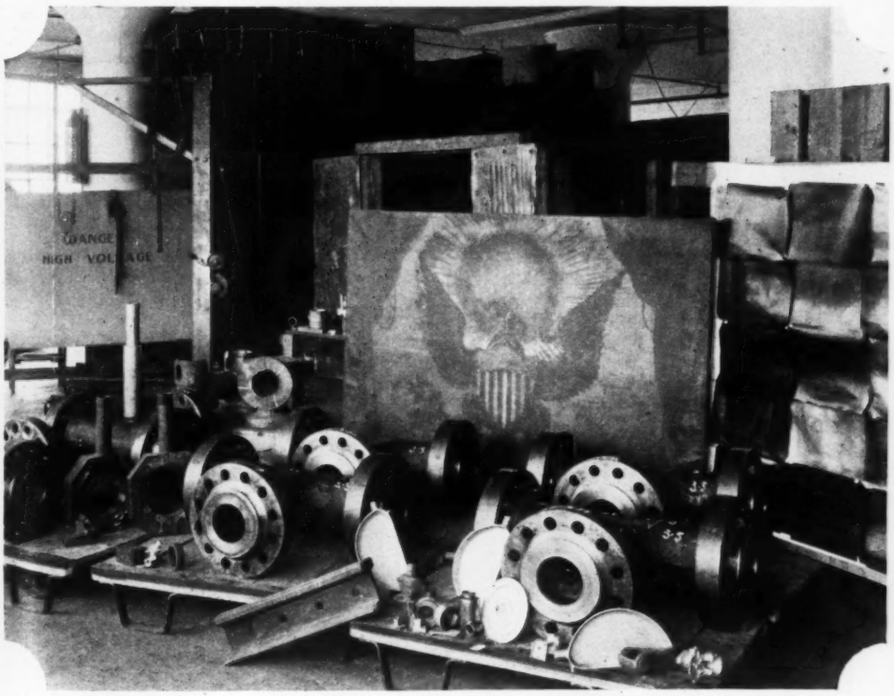


FIG. 4—OTHER FITTINGS TESTED INCLUDED SIX-INCH TEES, REQUIRING NINETEEN PICTURES EACH

What the Tests Show

X-ray tests, confirmed by sectioning or other collateral analyses, indicate that undesirable internal conditions in castings fall into relatively few classes, all traceable to definite and simple causes. Most of these causes, if not all of them, can be eliminated by proper foundry practice. Experience shows that when chronic defects have been corrected by making the indicated changes in casting technique, they tend to stay corrected. It is possible thus to eliminate from 75 per cent to 90 per cent of the major defects.

The principal undesirable conditions are:

Much valuable information about X-ray inspection is contained in "Recommended Practice for Radiographic Testing of Metal Castings." Designation: E. 15-29, published recently by the *American Society for Testing Materials.*

1. Gas, slag and sand pockets due to loose dirt in the mold.
2. Gas cavities due to imperfectly deoxidized metal.
3. Sand inclusions due to cutting of mold or runners.
4. Pipe or primary shrinkage caused by failure of risers to function as intended.
5. Secondary pipes caused by flow of viscous metal through constricted channels in the casting during the final stages of solidification.
6. Shrinkage cracks starting from a sinus-like cavity developed during cooling.
7. Rupture developed during hydrostatic test.

These voids, inclusions and all other variations in the density or thickness of the metal of course affect the exposures. They are all clearly defined on the film, the interpretation of which is an art in itself. Accurately done, however, it is possible to determine definitely the defect and, in many instances, castings may be saved by cutting out the defective part and replacing it by the simple expedient of welding. The depth of the cavity, which is often veneered when taken from the mold, may be determined by means of special technique and exposure conditions. The most frequent and common defect found is that caused by shrinkage.

Curing Hay Fever with Controlled Weather

By R. F. Morrison

NO ONE who has ever suffered from it enjoys the prospect of another attack of hay fever. Yet, where pollens are the cause, it comes to many with frightful regularity. The tree pollens are in the air from February to June, the grass pollens from May to August, and the weed pollens from August till frost appears. Out of this eight-month period, at least three or four months bring little pleasure to many people. A portion of these are subject to a perennial form of the disease. This consists not only of the pollen forms of hay fever but also the irritations set up by some foods.

Another form of hay fever has come into existence with the advent of civilization. In this case mechanical, chemical, odorific and thermal dusts, fumes and conditions are the primary causes of irritation. The dusts include sweepings, soaps, powders, lye and face powders. Fumes and odors include ammonia, scented perfumes, musty air and stable odors. Temperature changes and drafts complete the list. These irritants are not seasonal and may bring on a case of hay fever at any time.

It is evident, from the causes which have been named, that hay fever can be said to be due almost entirely to atmospheric irritants.

The circumstance which makes these irritants so troublesome to certain persons is an abnormal nose or throat in which the mucous membrane is extremely sensitive. This could be a chronic catarrhal infection or some physical deflection of the cartilage. The latter could be the cause of frequent contact between various exposed portions of the mucous membrane. In the latter case, where hay fever is concerned, operations have not always proved as successful as the elimination of the irritant.

Present methods of treatment consist of inoculation, over a 10 to 15 week period, prior to the time at which the irritant generally begins to affect the patient. Tests are first made to determine the susceptibility of the patient to various irritants and thus learn just what irritant is behind the trouble. Where the disease does not respond to treatment (which usually consists of dark glasses, boric acid eye-wash, adrenalin nasal sprays and saline douches) high altitudes or ocean voyages are recommended, since they afford freedom from pollens.

The cause for hay fever is due, as has been seen, to atmospheric irritants and, as the present methods of treatment are largely preventive it seems logical to conclude that controlled weather affords a reasonable means of treating hay fever. Washed air, in a hospital, would eliminate all dangers from irritants in the nature of dusts, fumes and, to a considerable extent, odors. The temperature would be uniform. Drafts would give place to fixed air motion which, at low speeds, is hardly noticeable. Best of all the temperature, humidity and air motion could be adjusted to give real human comfort as far as atmospheric conditions are concerned. This latter feature would mean much in preventing fevers and in maintaining normal conditions as to the nerves.

Where such treatment for hay fever is provided in hospitals, a certain amount of experimentation, now needed in this disease, could be attempted. This would be the determination of the best relative humidity for healing the mucous membrane. There is plenty of evidence at hand, at present, to show that the relative humidity does have an effect. To what extent is it helpful and how much is needed?

The air motion is not likely to exceed 100 feet per minute and may be as low as 50 feet per minute. For either speed it is evident that human comfort will require some variation of air temperature (the temperature of the dry bulb) and probably even relative humidity for summer and winter weather. In order that the various possible human comfort combinations may be understood, Table 1 has been prepared from the human comfort data of the research laboratories of the *American Society of Heating and Ventilating Engineers* and the *Harvard School of Public Health.*

TABLE 1—COMBINATIONS WHICH GIVE BEST HUMAN COMFORT

	Dry Bulb Temp. F.	50 F. P. M.		100 F. P. M.	
		Wet Bulb Temp. F.	Rel. Hum. %	Wet Bulb Temp. F.	Rel. Hum. %
For Summer	72	72	100		
	74	69	78	72	89
	76	66	60	68	66
	78	63	43	65	49
	80	60	30	62	35
For Winter	70	62	64	64	72
	72	57	40	60	50
	74	52	18	55	28
	76			49	7

These data are for persons normally clothed, slightly active and in health. Furthermore, they give the maximum comfort for rooms where the persons who use them move in and out frequently. While there is no experimental evidence comparable to this information that covers the situation where human beings are not well, and while there can be no doubt that people who suffer from nose, throat and lung diseases are in need of higher temperatures than those who are healthy, yet this data is so reliable that it will need very little alteration to meet particular needs such as are found in hospitals.

Referring to the table it will be evident that each combination shown gives considerable variation in temperature, air motion and humidity. All of these combinations are equally comfortable. Furthermore, there are other combinations of equal comfort which can be built up out of the table presented. The purpose of the table is readily seen by the following example: A dry bulb of 74, wet bulb of 69 and relative humidity of 78 per cent in the summer is equally as comfortable as a dry bulb of 80, a wet bulb of 62 and a relative humidity of 35 per cent in summer even though the first condition is for an air motion of 50 feet per minute and the second for 100. Furthermore, both of these conditions, in the summer, will give equal comfort with any condition in the table for winter where the latter condition is used in the winter. This will be true only where people move in and out of the conditioned room and not where there is practically no movement from the room into the outside air.

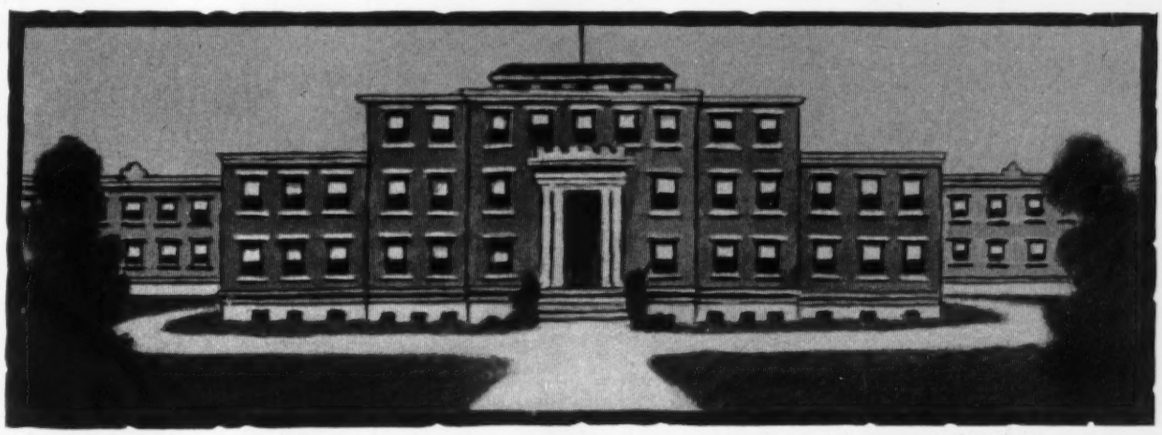
It may be interesting to add a bit of actual experience. Hay fever, as can be readily seen, often is due to physical conditions similar to those had in various forms of sinus diseases. Many specialists send patients with the latter trouble to high altitudes. At the same time numbers of those with sinus troubles have been greatly benefitted by living at the seashore. In both cases the pure air appears to be one of the benefits obtained by the change. At the same time there can be no doubt that the higher humidity of the ocean beach, added to the small amount of salt in the air, has a healing effect that is not obtained in the mountains. From this it appears evident that hay fever, and sinus diseases as well, might be cured by air conditioning at higher humidities where the air is permitted to become the carrier of a predetermined amount of salt or some other healing chemical.

Transporting Pipe in Alps

HYDRO-ELECTRIC installations are now in course of construction in Switzerland, near the Grimsel Pass which unites the Rhone Valley with the Valley of the Aare, a tributary of the Rhine; these will, when completed, be the most powerful on the continent of Europe and will develop a total of 360,000 horsepower. The different installations, reservoirs, conduits, power

stations, etc., will extend for 20 km. (about 12½ miles) and will utilize a fall of 1,200 m (3,936 ft.) at different levels.

In order to convey materials to the spot, a funicular has been built, the ramp of which, 105% (i.e., an incline of 46½°), is the steepest in Europe. It is intended for the transport of materials up to 10 tons in weight, and ascends from the Handeck to the Gelmersee plateau, a difference in height of 450 m (1,476.45 ft.) in a distance of 1,000 m. (3,281 ft.) only. The carriages are furnished with quickly-acting brakes which make it possible to stop dead in case of accident after 30 cm. (about 12 inches) run only.—*Swiss Industry and Trade.*



INTERVIEWS OF INTEREST

Prevention of Dusts is Vital in Mines

Silicosis, as one of the most dangerous of our industrial diseases, has received attention both as to cause and control. It was formerly thought that it was caused by irritation from large particles of silica dust inhaled into the lung. Recently, however, it has been demonstrated that it is the finer particles which do the damage and that these fine dusts seen to obey the laws

FIG. 1—WET DRILLING

FIG. 2—USE OF WATER WHEN COLLARING

of gases rather than those of solids. In metal mining, dust particles of less than 10 microns are the chief hazard, while dusts, such as those produced by blasts, are dangerous when they are less than 2 microns.

The problem is to prevent the dust from permeating air which the miners breathe. Preventive methods which are most effective "include wet mining methods—wet drilling and wetting sides, roof and bottom, muck, and rock piles; the use of sprays and water blasts to lay the dust after blasting; good mechanical ventilation to replace dusty air with clean air; and physical examination of all miners before employment and periodically thereafter," states Dr. R. R. Sayers, chief surgeon, Bureau of Mines.

Mine ventilation means many things, depending on the individual case; it may include the factors of dust, heat, barometric pressure, foreign gases, humidity. The air conditioning must then be quantitative; it must eliminate dust, and it must be prepared to prevent disaster caused by gas.

The amount of fresh air to be supplied to the mine varies with the amount of dust present, "but when two machine drills of the type which produce the least dust, are being used, 800 cubic feet of clean air must be delivered per minute at the working face to reduce adequately the amount of dust." Ventilation by compressed air alone has not always proved satisfactory. A thorough system of large surface-fans supplemented by underground auxiliary fans, and supplemented by air shafts controlled by doors or canvas tubing or galvanized iron pipes to guide the air, is equally imperative.

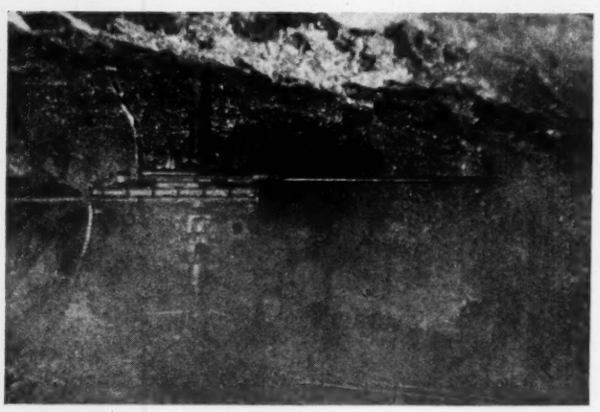


FIG. 3—DUST FORMED BY ACTION OF DRILL WHEN WATER FAILS FOR THIRTY SECONDS

Although the customary increase in heat in mines is 1 degree for every 70 feet of depth, this is not uniform. In Nevada there are mines which increase 1 degree for every 50 feet of depth and in South Africa others which increase 1 degree for every 200 feet. Nearly all mines have high humidity at great depths. Control of temperature and moisture are customarily done by introducing conditioned air, if possible; frequently the conditioning must be done both at the surface and underground near the working face.

Wetting surfaces at the working face is an efficient method of preventing dust. The illustrations shown here demonstrate the difference between drilling with water used in connection with the drill and without. In two of the pictures water is used; in the third, water has stopped for thirty seconds and the dust is so thick that the man at the left of the picture can barely be discerned.

Mechanical Equipment in Schools

Dr. O. F. Ball, president of The Nation's Schools Publishing Co., Chicago, stresses the importance of consistent and persistent study of school situations and holds that such preparedness is the only basis of obtaining a reliable picture of the changing standards of school building construction.

Dr. Ball has recently sponsored an analytical review of 440 school building projects based upon returns from superintendents and architects of the schools in question. New projects were included in the study along with smaller additions to older plants. For this reason, states Dr. Ball, it has been difficult to reduce all similar items to a comparable basis. The per cubic foot cost of construction revealed unexpectedly wide variations. The highest cost reported was 55c per cubic foot. This particular cost item has been reported only on the basis of actual figures given by architect or superintendent as only 193 of the 440 projects included the figures on cubical content.

Dr. Ball then discussed the current trends:

"Our picture of present school building activities indicates that an approximate 31 per cent of building construction relates to city high schools. City elementary schools are looking up. They command 35 per cent of present building emphasis. City junior high schools account for a little more than 9 per cent of our reports. Consolidated high schools are a little more than 9 per cent; consolidated elementary schools 2.5 per cent. Less than 4 per cent of our data have emanated from parochial schools, and a scant 11 per cent could not be classified.

"On the mechanical side, 33 per cent of schools are equipped with cafeterias; 35 per cent have kitchens; 62.05 per cent have boiler rooms; 32 per cent have fan rooms. A survey of heating systems indicate steam is widely used. This applies to 85.45 per cent of the schools. It was found 2.73 per cent have hot water heating systems, and about 9 per cent are heated with warm air systems.

"Low pressure boilers are employed in 65 per cent of buildings, with high pressure boilers in 10 per cent. More than 10 per cent of buildings are using oil burners, and mechanical stokers are employed in somewhat less than 10 per cent. The series under consideration employed ventilation systems in the following proportions: Fan and duct, 170, or 38.64 per cent; unit heaters, 191, or 42.41 per cent; and air washers, 23, or 5.23 per cent."

In certain sections, notably New York City, where school construction costs have increased spectacularly during recent months, the demand is growing for publication of separate cost figures for building materials and labor. It is with some interest, therefore, that Dr. Ball is responsible also for the publication of such additional data in a price index of school building and school supplies.

With 1926 representing 100, basic materials of twenty-two types that enter into schools have shown a reduction as from 133.42 in 1920 to 94.08 in 1929. Labor cost index was 79.30 in 1920, and 103.33 in 1929. The over-all reduction in price index for the same period was from 110.63 in 1920 to 97.97 in 1929.

Industrial Museum Achieves Engineering Perspectives

A museum of science and industry which enables the display of backgrounds of science alongside of the newest and best engineering solutions in modern society, and that without ignoring the social and cultural aspects of science and technology, is the object contemplated by Julius Rosenwald in giving Chicago the Museum of Science and Industry. It will reveal the technical ascent of man.

"A mere repository of historic models is not enough for museum purposes," states Waldemar Kaempffert, director of the Museum organization activities. "Neither do paintings and charts and mathematical physics serve alone to encourage engineering progress. Least of all is it possible to preserve the continuity of engineering thought if the machine last framed obliterates all models from which it sprang. We are always interested in the progressive principles of its evolution."

The Museum of Science and Industry will exhibit engineering in its developmental aspects. It will stimulate interest. It will answer inquiry. It will suggest new lines of departure and initiate new quests. "Co-operation on the part of engineering groups is essential throughout," states Mr. Kaempffert. "We must first look to engineers and to pioneer manufacturers of engineering products for information of what models are available, and where. We desire to be informed directly what these models are. Many interesting things were seen in Philadelphia on the occasion of the first international exposition of the *American Society of Heating and Ventilating Engineers* in January. We'll search for more and rescue from oblivion important engineering models outlived and now ignored."

It is of interest to observe that the first engineering committee to act as advisers in the work of organizing Chicago's industrial museum includes M. E. Cooley, dean emeritus of the College of Engineering and Agriculture, University of Michigan; E. L. Ryerson, of Joseph T. Ryerson and Son, Chicago; and Joseph W. Roe, professor of industrial engineering, New York University. These men were appointed by the *American Engineering Council.*

"Many exhibits in the Museum will relate to heating and ventilating," states Mr. Kaempffert. "We could fill an entire hall with ventilators alone. Ventilating engineers alone can pass upon these matters.

"At least one hundred engineering problems are before us now. The *American Engineering Council* will cooperate. Representatives of other organizations, too; for we serve not education merely but undertake to find true background stuff of uninterrupted, substantial progress."

The Museum of Science and Industry has been preceded by similar institutes in Munich, Vienna, Paris. The American technical museum will be housed in a $5,000,000 building. It will be ready to set in motion by 1933. It will never be really completed, for the task it undertakes is endless to keep all types of industry represented and to furnish a measure of their effect upon human kind.—*S. P. M.*

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

Sizing Pressure Regulators

"Pressure Reducing Valves," *by Sabin Crocker.*, Page 20, issue of January, 1930

WE recently read the article in HEATING, PIPING AND AIR CONDITIONING for January on the selection of pressure reducing valves.

This article is very clear and explanatory insofar as the construction and operation of the valves are concerned, but we do not agree with the author's contention in the statement that from the similarity of flow conditions between safety valves and pressure regulators, it is apparent that if this formula applies to one it should to the other.

As pressure reducing valve manufacturers, we wish to state that there is no comparison in the flow of steam from a safety valve and the flow of steam from a pressure reducing valve.

On a safety valve, the flow is from pressure to atmosphere and the velocities may run anywhere from fifteen to thirty thousand feet a minute. The usual safety valve, when in operation, creates a considerable amount of noise, which is one of the salient points in the selection and sizing of pressure reducing valves. A reducing valve must be of sufficient size to provide all of the steam required and at the same time be as noiseless as possible.

Naturally, in sizing a pressure reducing valve, the velocity of steam passing through the valve is determined by the engineer designing the job. Any formula which does not tie down the velocity of the flow of steam through the valve is absolutely worthless.

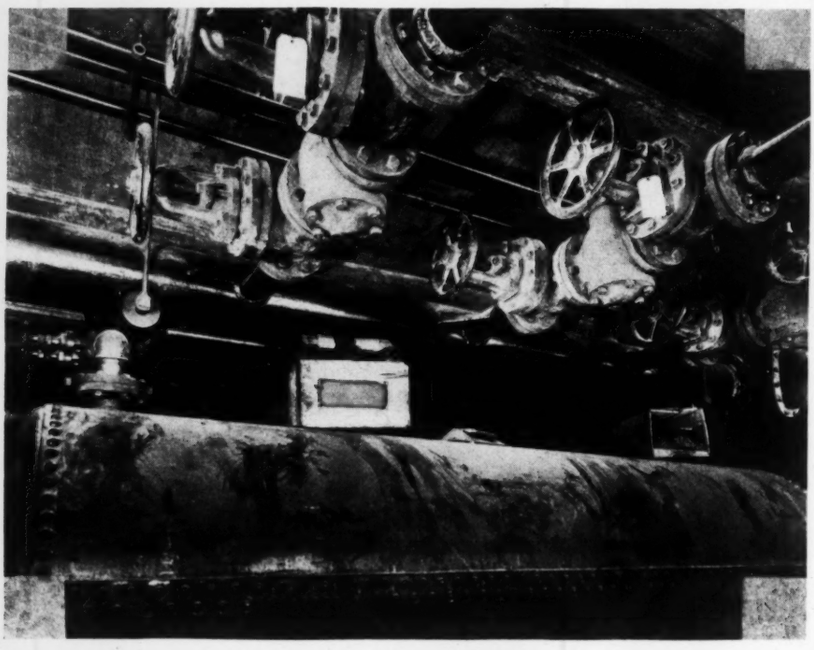
We believe that this part of the article is extremely misleading, as the velocity of steam has a great deal to do with the size of the valve. Naturally, with this formula a 3-in. valve would be sufficient to perform the service, where a reputable valve manufacturer would in all probability require a 6-in. or 8-in. reducing valve. —*C. H. Dutcher, Kieley & Mueller, Inc.*

The Author's Reply

The following comment is offered in reply to points raised in the above letter:

Apparently, your correspondent has misunderstood or misconstrued statements in the writer's article on pressure reducing valves regarding flow conditions in safety valves and pressure reducing valves. A brief review of steam flow principles may be of assistance in making the intent of this part of the article more clear.

To avoid unnecessary complications, this explanation will be confined to conditions where the discharge pressure does not exceed 0.6 the initial pressure, thus keeping within flow conditions where Napier's formula applies. According to generally accepted principles for saturated steam flow, the discharge through an orifice depends on the initial pressure and the area of the orifice, and is independent of the back pressure, provided the latter does not exceed 0.6 the initial absolute pressure. This being the case, the flow for any initial pressure and size of orifice will be essentially the same irrespective of



A BATTERY OF REDUCING VALVES

whether discharge is to the atmosphere or into a pipe line.

Consider, for example, a 5-inch valve with a 45 deg. bevel seat and flow conditions as follows: 10,000 lb. of saturated steam per hour at an initial pressure of 100 lb. gage; velocity in 5-inch pipe approaching valve approximately 4,600 ft. per min.; reduced pressure not to exceed 50 lb. gage. The lift of the disc required to give a discharge of 10,000 lb. per hour is computed from the Boiler Code Formula as follows: (See my original article.)

$W = 110\ PDL$ or $L = W/110\ PD$

$L = 10{,}000/(110 \times 115 \times 5) = 0.158$ inches

The orifice area corresponding to this lift is

$A = 3.1416\ DL$ sine of seat angle

$= 3.1416 \times 5 \times 0.158 \times 0.707 = 1.76$ sq. in. $=$ 0.0122 sq. ft.

The specific volume of steam at 100 lb. gage is 3.88 cu. ft. per lb. and the velocity entering the orifice is

$$V = \frac{10{,}000 \times 3.88}{60 \times 0.0122} = 53{,}000 \text{ ft. per minute}$$

The same flow conditions will exist through the port of either a safety valve or a pressure regulator irrespective of whether the discharge pressure is 50 lb. or 5 lb. In order to satisfy the laws of steam flow, the orifice area through the valve port usually is approximately 1/6 to 1/8 the area of the passage through the valve body. Consequently, velocity through the port must be correspondingly higher than through the valve body. In the case of the pressure regulator, it would be desirable, of course, to increase the size of the discharge piping close to the valve so as to obtain a reasonable velocity after the steam has assumed the increased volume corresponding to the lower pressure. At a reduced pressure of 50 lb. gage, a 6-inch pipe would be required to give a velocity of approximately 6,000 ft. per minute. At a reduced pressure of 5 lb. gage, a 10-inch pipe would be required to give the same velocity.

The table given below furnishes data in support of the writer's statement that the basic principles of steam flow apply to both safety valves and pressure regulators.

Initial Pressure, Lb. Gage	Reduced Pressure, Lb. Gage	Lb. of Steam Per Hour	Size Safety Valve, Inches		Size Pressure Regulator, Inches**		
			Ordinary*	High-Lift	I	II	III
200	50	10,000	3½	2	2½	3½	4
200	5	10,000	3½	2	2½	3½	4
100	50	10,000	5	3	3	4½	5
100	5	10,000	5	3	3	4½	5
50	5	10,000	2—5	4	5	6	8
20	5	10,000	4—4½	6	8	8	.....

NOTE *: From pamphlet "Safety Valve Capacity," by Phillip G. Darling. See also paper by same author, A.S.M.E. Transactions 1909, page 109.

NOTE **: From manufacturers' catalogues.

Examination of the above data indicates a marked tendency for agreement in size between pressure regulators and safety valves for the same flow conditions. While very high velocities must obtain through valve ports, as shown by the formulas concerning flow through orifices, this does not mean that velocities need be anywhere near so high in the piping on either side of the valve. The use of a larger pressure regulator with a larger diameter valve port, as implied by your correspondent, merely results in reducing the lift required to produce a given steam flow—thus maintaining the orifice area and velocity through the port the same as it was before.

If I am wrong in my analysis of the situation, I should be glad to hear further from your correspondent after he has had opportunity to examine this reply.—*Sabin Crocker.*

Mr. Dutcher's Second Letter

In reply to the above:

The discrepancy that we pointed out in our previous letter still exists in Mr. Crocker's computation as submitted in his answer to the point which we have taken.

The point that we raise is that there is a great difference in the operation of a pressure reducing valve and a safety valve. This fact is borne out more clearly by the figures submitted by Mr. Crocker. In other words, in the computation submitted the lift of the disc in the safety valve is taken as 0.182 in. for a 5-in. valve.

In computing this for the capacity of steam at a certain volume, we have a velocity through the valve of 46,000 feet per minute. Now, the point that we bring up is the difference in computing a safety valve and a pressure regulator.

Take an ordinary 5-in. balanced regulating valve which would have two 3½-in. discs. The normal lift on a pressure regulating valve is one-fourth the nominal diameter of the pipe. In other words, a 5-in. pipe—1¼-in. lift. Two discs, ⅝-in. lift for each disc.

Figuring back to 3½-in. discs with a ⅝-in. lift, we get an area through the valve discs with the valve wide open for maximum capacity of .096. Figuring back for velocity to pass 10,000 lb. of steam at 100 lb. pressure we get

$$V = \frac{10{,}000 \times 3.88}{60 \times .096}$$

which gives a velocity of approximately 6,750 ft. per minute. This velocity is accepted as being practical and eliminates the possibility of noise in the operation of the valve.

Of course, it is understood that particularly heating loads in buildings are figured for a temperature difference of 70 F—that is, zero outside, 70 F inside. Naturally this is the maximum load and generally the load runs more to about 50 per cent of the maximum condition.

We, therefore, find that any valve figured in this way with a velocity of 6,000 ft. a minute permits quiet action and very safe and conservative velocities through the valve for normal operation. Bear in mind that we do not question this article except on this one particular point.—*C. H. Dutcher, Kieley & Mueller, Inc.*

Mr. Crocker's Second Letter

The following is offered in reply to Mr. C. H. Dutcher's second letter:

In discussing the action of the 5-inch pressure regulator used for illustration in my first letter, Mr. Dutcher gives the normal lift of a single seated valve as ¼ the nominal size, or presumably ¼ the diameter of the opening through the port. This relation corresponds to an orifice area equal to the nominal cross-sectional area of the pipe, since if d is the nominal diameter of the pipe and the actual diameter of the valve seat:

Area of pipe $= \pi d^2/4$

Area of orifice with flat seat = lift × circumference of seat $= d/4 \times \pi d = \pi d^2/4$

If the valve has a 45 deg. bevel seat, the area of orifice becomes

Area of orifice = sin 45° × lift × circumference of seat $= 0.707 \pi d \times$ lift

If the area of the orifice is to be equal to the area of the pipe, then

$$\pi d^2/4 = 0.707 \pi d \times \text{lift}$$

and the lift for full opening is

$$\text{Lift} = \frac{\pi d^2}{4 \times 0.707 \pi d} = d/2.828$$

In the case of a double ported valve, Mr. Dutcher has assumed two 3½-inch ports. The opening through two 3½-inch ports is essentially the same as through one 5-inch port, but the lift to give full area with a flat seat is 3½ ÷ 4 = ⅞ inch rather than the ⅝-inch lift assumed by him. In the case of a 45 deg. bevel seat the lift would have to be 3½ ÷ 2.828 = 1¼ inches to obtain full area through a 3½-inch seat.

A regulator having a port area equal to the normal area of the steam piping offers no more resistance to steam flow than would a wide open globe valve—hence in the wide open position it could not possibly serve as a means for reducing pressure from 100 to 50 lb. gage to fulfill the conditions given in my example.

In his second letter Mr. Dutcher applies a double ported valve with two 3½-inch seats having a valve lift of ⅝ inch to the example given in my first letter. Apparently Mr. Dutcher intends to imply that this arrangement will give a flow of 10,000 lb. per hour from an initial pressure of 100 lb. gage to a reduced pressure of 50 lb. gage. This cannot be true, however, since from Napier's formula applied to a 45 deg. bevel seat

$$W = 110\ PDL$$

or for a double ported valve

$$W = 2 \times 110\ PDL$$

$= 2 \times 110 \times 115 \times 3.5 \times 0.625 = 55{,}200$ lb. per hour.

It is apparent from the above that two 3½-inch ports with a valve lift of ⅝ inch will give a steam flow of 55,200 lb. per hour rather than the 10,000 lb. per hour of my example. The corresponding velocity is arrived at as follows:

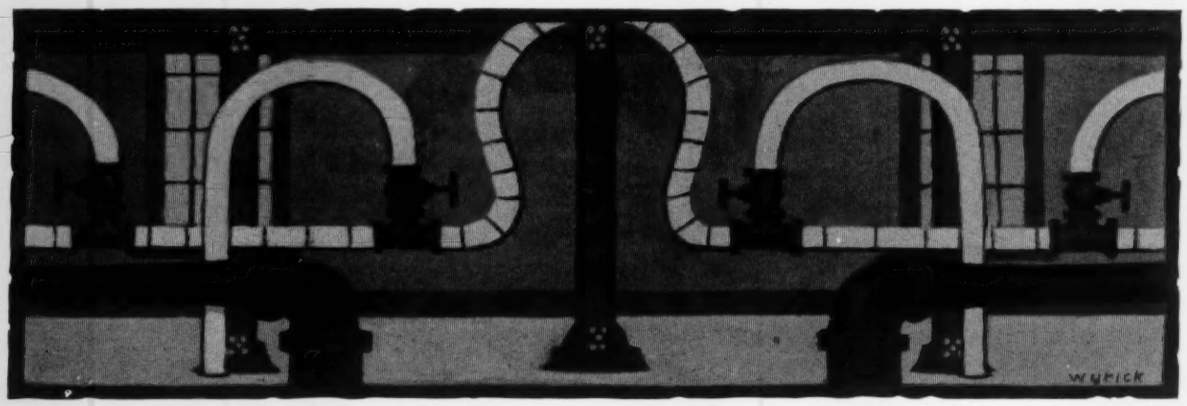
Area for 2 ports $= 2 \times 0.707 \times 0.625 \times \pi\ 3.5 = 9.73$ sq. in.

$= 9.73 \div 144 = 0.0675$ sq. ft.

and the velocity entering the ports will be

$$V = \frac{55{,}200 \times 3.88}{60 \times 0.0675} = 53{,}000 \text{ ft. per minute,}$$

which, as would be expected, checks the velocity computed in my first letter. According to the laws of steam flow, it is impossible to decrease the velocity through the regulator port, for any given set of flow conditions, by increasing the lift or diameter of the seat. To increase the lift or diameter merely increases the rate of steam flow without decreasing the velocity. That velocity depends on initial steam conditions alone, provided the back pressure does not exceed 0.6 the initial pressure, is demonstrated readily by substituting in the previously



mentioned formula for 45 deg. bevel seated valves, $W = 110\ PDL$:

$$A = \pi\ DL\ 0.707 \qquad \text{and} \qquad A = \frac{144\ WS}{60\ V}$$

combining and solving for DL

$$DL = \frac{144\ WS}{60\ V\ \pi\ 0.707}$$

Substituting in the original equation $W = 110\ PDL$

$$W = \frac{110\ P\ 144\ WS}{60\ V\ \pi\ 0.707}$$

combining and solving for V

$$V = 119\ PS \qquad \text{(45 deg. bevel seat).}$$

The above derivation demonstrates that velocity through the seat depends on initial pressure and specific volume alone. As a check on the accuracy of the result, refer again to the previous example

$V = 119 \times 115 \times 3.88 = 53{,}000$ ft. per minute

That the velocity entering the port does not vary much over a wide range of initial pressures is shown by applying the $V = 119\ PS$ formula to the following examples:

Initial Pressure Lb. Gage	Specific Volume Cu. Ft. per Lb.	Velocity through Port Feet per Minute
200	2.14	54,700
100	3.88	53,000
20	11.89	49,500

In concluding, I would again call Mr. Dutcher's attention to the table of pressure regulator capacities given in my first letter as evidence that actual regulator sizes are not so different from safety valve sizes. Apparently Mr. Dutcher is under the impression that pressure regulators are noiseless in operation. This is far from being the case, and I should appreciate having an opportunity to take him around to *hear* some as they "whistle about their work" in Detroit plants. It is not my intention to imply that pressure regulators make as much noise as safety valves. The huddling rings used to control the blow-back on safety valves, and the acoustic conditions set up by discharge to atmosphere, have a decided effect on the noise produced. In attempting to describe and compare the sounds of both, I should say that the noise made by a safety valve is a "roar," while "whistle" better describes that of a pressure regulator. The amount of whistle produced by a pressure regulator increases with rate of flow through the valve, and apparently is louder at high pressures than at low.—*Sabin Crocker.*

A. S. H. V. E. Code of Minimum Requirements for Heating and Ventilation of Buildings

By Samuel R. Lewis

"The Code of Minimum Requirements," *published by the American Society of Heating and Ventilating Engineers.*

In the March Heating, Piping and Air Conditioning, *Samuel R. Lewis discussed the first three sections of the American Society of Heating and Ventilating Engineers' Code of Minimum Requirements. Mr. Lewis took up the fourth, fifth, sixth, and seventh sections of the Code in April; this month, he comments on Sections VIII-X.*

Appearing with Mr. Lewis' comments in the March number was a brief description of the Code of Minimum Requirements.

Section VIII: Minimum Requirements for Pipe Sizes of Low Pressure Steam, Vapor and Vacuum Heating Systems.

The definitions of the various types of steam circulating systems, like most things on earth, are not perfect. They have been improved in the Guide 1930.

The entire trade nomenclature concerning steam heating systems is unfortunate. It is to be hoped that a move to improve the terminology will be made.

All these systems, for instance, use steam, and the systems which are called vacuum systems as well as those which are called vapor systems may be gravity systems. Those which are called vapor systems may often operate for long periods below atmospheric pressure.

There are parts of one-pipe systems in which the condensation and the steam flow in the same direction in the same pipe, and there are parts of two-pipe systems in which the steam and the condensation flow in opposite directions in the same pipe.

Can anything be clumsier than the term "down feed riser?" When is a trap a valve, and when is a valve a trap? Both terms are used indiscriminately in the trade parlance.

The Code in general is excellent in its disclosure of the basic mathematics. The pipe size tables have been simplified beyond those published in the Code, and the explanations of each table have been printed opposite each table in the Guide 1930. Some changes in the tables also have been made in the Guide, the result of further studies by the Research Laboratory.

All of the tables which use "square feet" might just as well be given in heat units, and I predict, now that a square foot isn't a square foot any more in any event, that this reform soon may materialize.

No one need fear to use the tables, as they are assuredly very conservative, based upon the experience and practice of many engineers. It is recommended, however, that the improved tables of the Guide 1930 be used in preference to these, and it is probable that a new Section VIII for the loose leaf folder of the Code of Minimum Requirements soon will be forthcoming.

Section IX: Minimum Requirements for Gravity and Forced Hot Water Heating Systems.

The terminology of this Code is superior to that used in the Steam Piping Code, and the fundamental data have been very thoroughly worked out. The charts look rather formidable to the uninitiated, but the well-worded examples are illuminating.

Hot water heating is adaptable to difficult situations and generally is satisfactory and long-lived. Because of the popularity of proprietary low pressure steam systems it does not always receive the attention warranted by its manifold virtues.

Section X: An excellent series of definitions is given, though item *C* mentions fresh air. We are trying to eliminate this term, since no one knows what is "fresh" air. The term "outside" air as mentioned alternatively is much to be preferred. The term "supply" covers what is mentioned in definition *C*.

Following definition *X*, Register, it would be helpful if it were made clear that the register always means the combination of a grille and a body, the register body being the operable damper and its box or frame.

Excellent suggestions are given for air duct construction, gages, etc. It might be helpful to add to requirement 8, that drains from air ducts should have brass screw plugs, and normally should be kept closed, since seals in traps quickly will evaporate.

An excellent scheme is to have air duct drainage provided by open gutter through a weep hole to some adjoining lower room, letting the seepage escape to the sewer through a floor drain in this room.

Many engineers call for a manhole door, or at least a handhole door, near every damper of whatever nature, in every air duct. This is because eventually dampers will work loose from their shafts or because doubt may arise as to the relation of damper blade and damper shaft. Often a damper may engage some ob-

struction and become bent on its shaft, no longer telling a true tale.

There is good logic in the idea that if we must have a damper we must provide for observation and maintenance of that damper.

The Code is weak in items 12 and 13 having to do with building-outlets and building-inlets.

The outlets of item 12 must not look out horizontally, even though they may be fan-delivered. A cold wind will throttle back even a high velocity fan-propelled current of warm air, and no dependence can be placed on such a ventilating system. The outlet must be above or away from the influence of surrounding structures, and while it may have a storm hood, the ordinary ventilator interposes considerable resistance to air delivery, and the nearer the outlet approaches an open upward-looking chimney, the better.

Item 13 should give other metals beside galvanized sheet steel a chance. No engineer will object, for instance, to the use of copper or monel metal or rustless steel if he can get it.

It seems to me that Item 13 should be rewritten, and that it should call very definitely for tight, durable, adjustable, easily-operated doors or shutters for every opening for supply of outside air. Whenever we find a ventilating system without such doors we find a wasteful ventilating system, unless it happens to be one which runs twenty-four hours every day.

Item 49 of Section X should be revised to include the results of the Armour Institute research, which indicates that if an accurate anemometer is used over symmetrically divided small areas of a register, and shall be held touching the register, the average velocity, multiplied by the gross area of the grille plus the net area of the holes through the grille, divided by 2, will give a truthful determination of the air volume.

Item 63-A is not clear in stating that the fan should be fastened solidly to the 4-in. thick wooden frame (a complete platform under the fan and its motor is better); the frame being floated on the felt or cork. Some doubt also is expressed as to whether ½-in. of felt is in any way to be compared with 2-in. of cork. Also, the platform will travel unless held in leash by insulation-edged, permanently-fastened curbs.

Item 66: If the engine is direct-connected to the fan, the fan foundation must by all means be integral with the engine foundation.

There is nothing consistent about a rule which prohibits speed adjustment with a steam engine, an ideal prime mover for variable speed, and which in the next paragraph advises electric motors with variable speed. Items 66 and 67 should be rewritten, I believe.

In general, adjustable speed for ventilating fans is not to be desired, though in special cases it is permissible.

Item 70 is inconsistent. No one has found less space in public buildings for fans than in private buildings, and there seems to be no good reason why a public building fan should not have a belt just as freely as may a private building installation. I do not think anyone knows how to insulate an electric motor against sound without insulating also the thing which the motor propels.

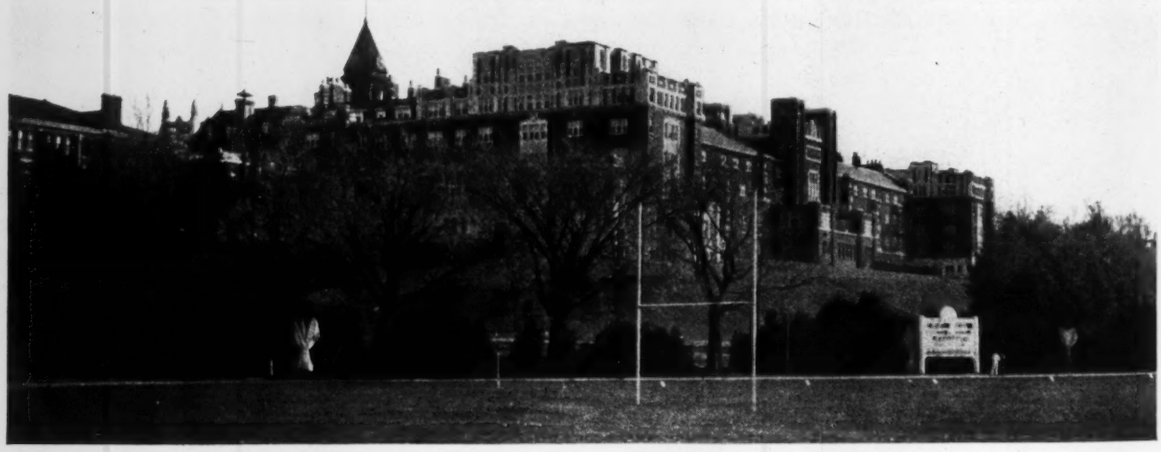
Usually, if we are to insulate a belted motor against sound, we must float both the motor and the thing which it propels on the same foundation, which must be rigid enough to maintain proper belt-tension.

Modern multiple rope and belt drives are too efficient and durable and valuable thus to be cavalierly condemned.

My impression is that Section X of the Code needs a bath.—*Samuel R. Lewis.*

Valves

Valves for distribution lines in rail-road yards should be of heavy construction, and also designed to be repaired quickly. The steam valves are subject to hard usage, are handled by all classes of labor and are allowed to wire draw which will cut valve seats very quickly on account of the high percentage of moisture that these lines carry. Valves for water, air and oil lines should also be of heavy construction and easily maintained. Valves and accessories for all distributing lines should be so installed that employes or persons walking through the yards are not likely to fall over them.



Men's Dormitory at Carleton College, Northfield, Minn. Holmes & Flinn, Architects. Samuel R. Lewis, Heating Engineer

Heating·Piping
and Air Conditioning

American Society *of* Heating *and* Ventilating Engineers

JOURNAL SECTION

MAY *1930*

Material in the Section prepared by Publication Committee: G. L. Larson, *Chairman*; H. H. Angus, and W. A. Rowe ; A. V. Hutchinson, Manager of Publications

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Effect of Oil Burner Adjustments on Overall Boiler Efficiency and a Progress Report on Research

By L. E. Seeley[1], New Haven, Conn.
NON-MEMBER

A study of the operation of domestic oil burners in boilers and furnaces is being conducted at Mason Laboratory, Sheffield Scientific School, Yale University, under a co-operative agreement with the Research Laboratory of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *and The American Oil Burner Association.*

In this investigation various factors affecting the operation of any type of burner in any type of heating system, are being studied. Such factors as the effect of the adjustment of the air-oil mixture and the draft carried in the furnace on efficiency of combustion and the combustion space necessary for various types of burners are included in the program.

It is hoped that the study will result in data of value in writing a code for testing boilers or furnaces and oil burner combinations. It is planned to present a series of short technical articles on the various phases of the study as they are completed. It is hoped that these short articles will encourage constructive criticism of the methods used and suggestions for improvement of the work.

OIL burners are built in many sizes to take care of every job from large to small. A few such burners of similar design, though different in size, usually suffice to give what may be called a line of burners. This implies that the burners can each handle amounts of oil ranging from some minimum to some maximum. These amounts are usually stated by the manufacturer in terms of the range in heating load that a burner can carry, or, what is more exact, the minimum and maximum amounts of oil that a given burner can properly handle.

Since the function of an oil burner is to burn oil correctly and at the proper time, it is only logical to expect that a burner shall consume either the maximum, minimum or any intermediate amount of oil with nearly equal facility. This merely means that the oil and air must easily be adjustable within the limits set and the burner will insure equally effective mixing and combustion of said oil and air.

A burner of the gun type, using a combination of air and pressure for oil atomization, was tried to ascertain whether it could meet the foregoing conditions. The air was supplied by a blower which was equipped with an air regulating device. The burner was set for about its maximum rate of oil burning and a gas analysis of the combustion products indicated good combustion. The amount of oil was subsequently reduced, but it soon developed that the air could not be reduced in amount with equal facility. The result was that the combustion became progressively poorer, due to excess air, as the rate of oil burning was decreased. It should be remembered that this is the best performance that could be obtained with the regulating devices as supplied. This burner was placed in a boiler and the efficiency of the boiler was obtained in each case.

The next step was to develop a method of better air regulation to endeavor to improve combustion. This was done by making a set of orifice plates which could be fastened to the fan inlet. If the burner was able to mix the air and oil properly, the combustion results should be improved. It was found possible by this means to get 10 per cent or more CO_2 without CO at the various rates of oil supply.

This demonstrated that the device as supplied by the manufacturer was not truly adapted to the range of capacities which it should have been, due to a poor method of air regulation. The importance of this is shown when the boiler efficiencies are compared. Over the range of boiler outputs obtained, the efficiency in the first series of tests due to poor combustion ranges from about 0 per cent to 15 per cent lower than it should when compared to the second series where combustion was maintained nearly constant in quality. Fig. 1 shows the results graphically.

The following deductions can be made:

1. A burner should be able to cover properly the range of capacities which the manufacturer claims. This means positive air and oil regulation and uniformly effective mixing of the two in the boiler furnace.

2. Even after a burner is so constructed that proper results are possible, some means should be provided to make these conditions probable when the burner is installed. It is highly probable that the correct adjustment will not be made unless the manufacturer provides a simple method of securing it. Unless the installer of an oil burner can tell the owner and the manufacturer how much oil the burner is consuming, and the quality of combustion, as shown by a flue gas analysis, he has failed to assure the owner that he is getting the proper amount of heat efficiently generated and has given the manufacturer no assurance that the burner is doing what it should to stay *sold* to the owner.

It should be pointed out that an ordinary installation

[1] Assistant Professor of Mechanical Engineering, Yale University, New Haven, Conn.

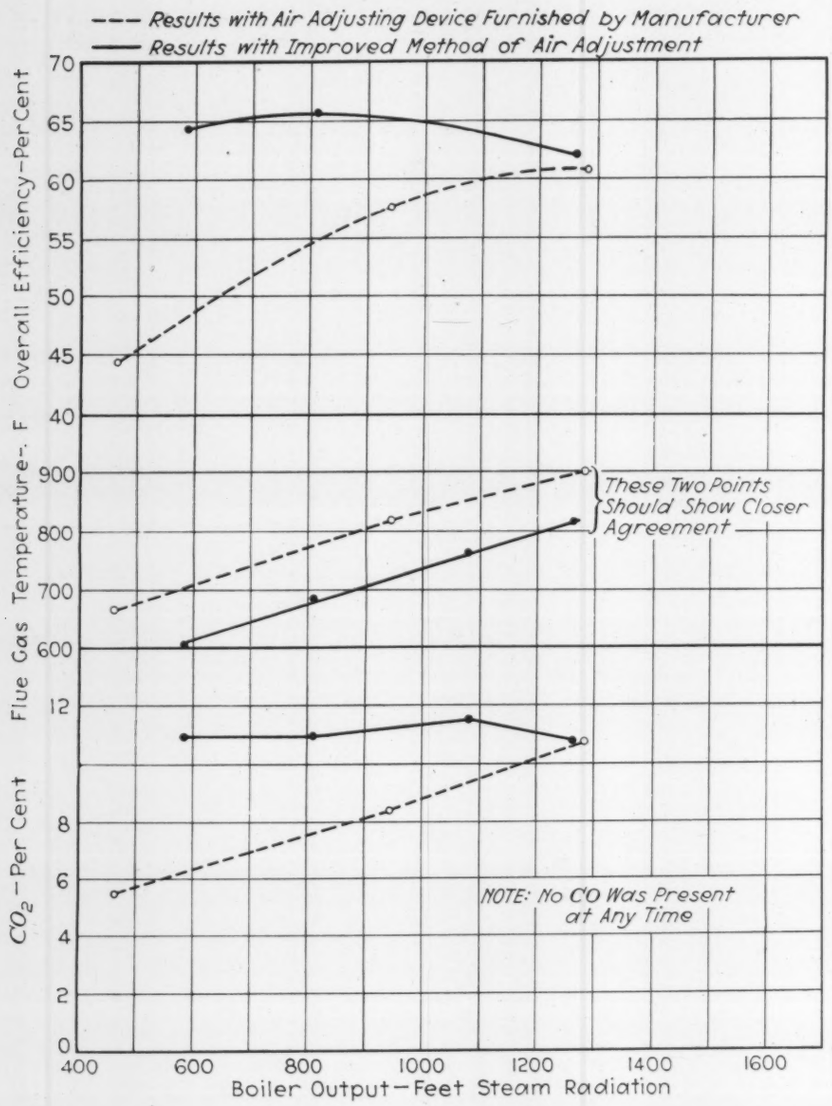

Results with Air Adjusting Device Furnished by Manufacturer
Results with Improved Method of Air Adjustment
Overall Efficiency-Per Cent
Flue Gas Temperature-° F
CO_2-Per Cent
These Two Points Should Show Closer Agreement
NOTE: No CO Was Present at Any Time
Boiler Output—Feet Steam Radiation

FIG. 1—PERFORMANCE CURVES OF OIL BURNER AND BOILER

may reduce the overall efficiency by 10 per cent and that this means about a 15 per cent increase in the owner's fuel bill. Oil burner manufacturers and dealers should fully appreciate the great importance of proper installation.

A continuation of the oil burner research indicates the progress of the work reported at this time in preliminary form, for the Annual Meeting of the American Oil Burner Association. The text of the report appears on the following pages and since it deals with the first and unfinished part of the program it should be viewed as a progress report.

Progress Report on Cooperative Research Program for Oil Burning Devices

THE following report prepared by L. E. Seeley[1] and F. J. Leonhard[2] gives:

1. The history of the present co-operative arrangement.
2. The character of work proposed.
3. The equipment being used.
4. The type of work being done at present, the reasons for this work and the leading conclusions obtained, and
5. A brief statement relative to future work and possibilities.

The need for an unbiased, practical study of domestic-size oil burners has been felt for some time. *The American Oil Burner Association* brought this matter to the attention of the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and a co-operative agreement was

[1] Assistant Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn., and Chairman of the Technical Advisory Committee on Oil Burning Devices of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

[2] Graduate Student, Mechanical Engineering Department, Sheffield Scientific School, Yale University.

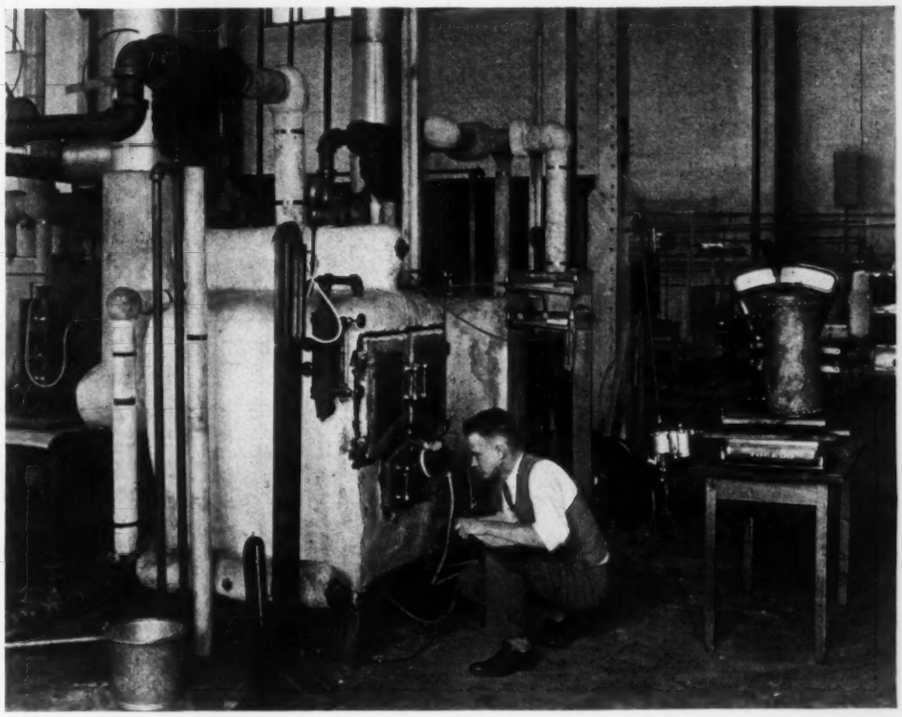


FIG. 1A — ARRANGEMENT OF BOILERS AND TESTING EQUIPMENT

arranged, the work to be done under a Society Technical Advisory Committee at Yale University, where excellent facilities were available.

The character of work proposed was to have a practical bearing on the everyday performance of oil burners. The work was not to be development work, but a study based on actual commercial burners. It was likewise decided that the boiler in which the burner is placed plays a very important part in practical operation. The task thus manifested itself as one dealing with actual commercial oil burners and boilers operating together under various conditions. The problem was to determine the important factors affecting the performance of this combination of devices (*i. e.* oil burner and boiler). It must be realized that the proposed tests in every individual case represent the combined behavior of two different pieces of equipment. By making the proper types of tests it should be possible by later comparisons to discover the influence of each device on their combined performance characteristics. This will be discussed in more detail later.

The equipment being tested has been generously loaned by various manufacturers. For obvious reasons, these devices are not being listed, described in detail or named. The oil burners are designated as *OB*-1, *OB*-2, etc., and the boilers as *B*-1, *B*-2, etc. Suffice it to say that five oil burners of representative types (*i. e.*, pot, rotary gun, etc.), and four types of boilers (*i. e.*, steel, cast-iron and copper-tube), are available for test. Each boiler smoke pipe is equipped with an automatic draft regulating damper. A steam separator is connected to each boiler and all other equipment necessary for boiler testing is available. It will save considerable time to omit test set-up details in this paper. Fig. 1*A* shows a general view of the arrangement of boilers and testing equipment. Each burner manufacturer installed his own burner. The service and co-operation received to date has been excellent. Fig. 2*A* shows how the details of furnace construction are recorded.

At present various combinations of boilers and burners are being tested under a number of operating conditions calculated to give a rather comprehensive picture of the performance characteristics of each combination. The present opinion is that these tests will afford certain logical comparisons of individual operating features after a sufficient number of combinations have been tested. The work is too incomplete at present to afford examples of the kind of comparisons previously mentioned.

Each boiler and oil burner combination is subjected to a series of tests which will be described. The reason for each type of test will also be explained and any leading conclusions pointed out. A complete set of tests contains seven series designated as Series *A, B, C, D, E, F* and *G*.

Series *A* is a set of tests run with No. 1 oil under definite operating conditions at various loads with continuous burner operation. This is designed to show the ability of the burner to operate at the maximum and minimum capacities which the manufacturer recommends for it. It also shows the ability of the boiler to absorb heat. Fig. 3*A* shows the economy curves for four boilers with four different oil burners. All that one may conclude from this is that the characteristic economy curves of boilers are not the same in amount or trend. Whether or not each boiler economy curve will retain its present shape with other kinds of oil burners is not known.

In order to compare the effect of a burner in various boilers it was necessary to fix certain operating conditions so it could be definitely stated that the burner was performing in essentially the same manner in each case. For example, it would be necessary to burn the oil at the same rates in each boiler and with the same quality of combustion as shown by a flue gas analysis.

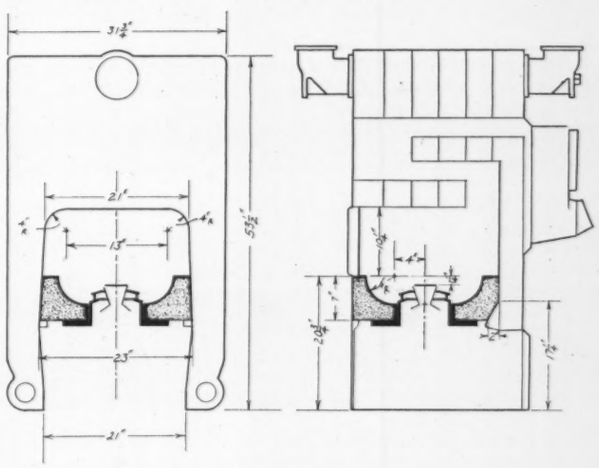
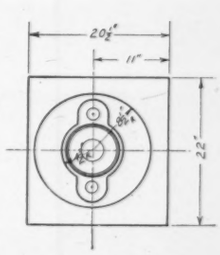


FIG. 2A—FURNACE INSTALLATION DIAGRAM

For this series the combustion is held to 10 per cent CO_2 if the burner can do it, or even if the burner can do better it is not allowed to. Ten per cent CO_2 without CO was chosen because it represents a reasonable ideal for practical work.



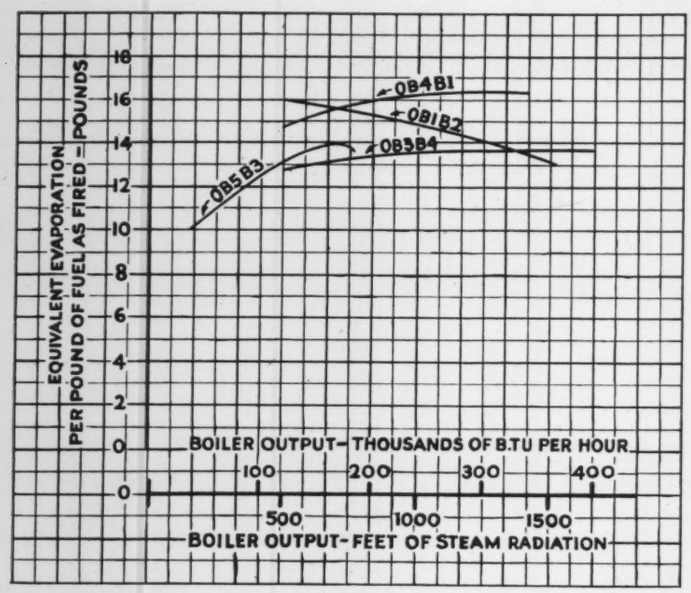


FIG. 3A—ECONOMY COMPARISON OF CONTINUANCE TESTS (SERIES *A*)

tion, *i. e.* 10 per cent CO_2. Any such change of draft represents a characteristic of the burner. Such a furnace draft may be criticized as being too low for practical work but is desirable as other test series demonstrate. Furthermore, a low draft tends to reduce air leakage and to maintain constant air leakage through the boiler setting at points not helpful to combustion.

Series *B* is a similar set of tests, using No. 3 oil. This is done only in case the manufacturer recommends No. 3 oil as being suitable for his burner.

Series *C* is a set of tests run at a boiler output of about 1000 ft. of radiation with the furnace draft increased progressively above the value specified by Series *A* up to 0.1 in. water. This shows the susceptibility of the oil burner to draft fluctuations. An output of 1000 ft. of radiation load was chosen because the boilers are rated thereabouts. Fig. 4*A* shows the effect of draft variation on boiler economy and on the combustion as shown by the percentage of CO_2. The one obvious conclusion is that the combustion

Since draft affects the combustion, it was standardized at 0.02 in. water in the furnace. The draft at the smokehood of the boilers will vary depending upon the boiler. That is a boiler characteristic. It is furnace draft only that affects

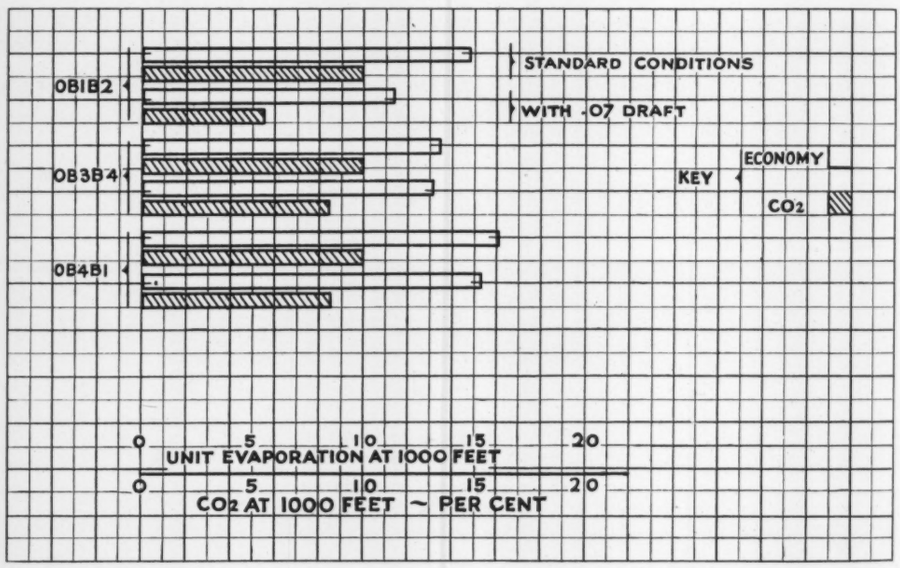
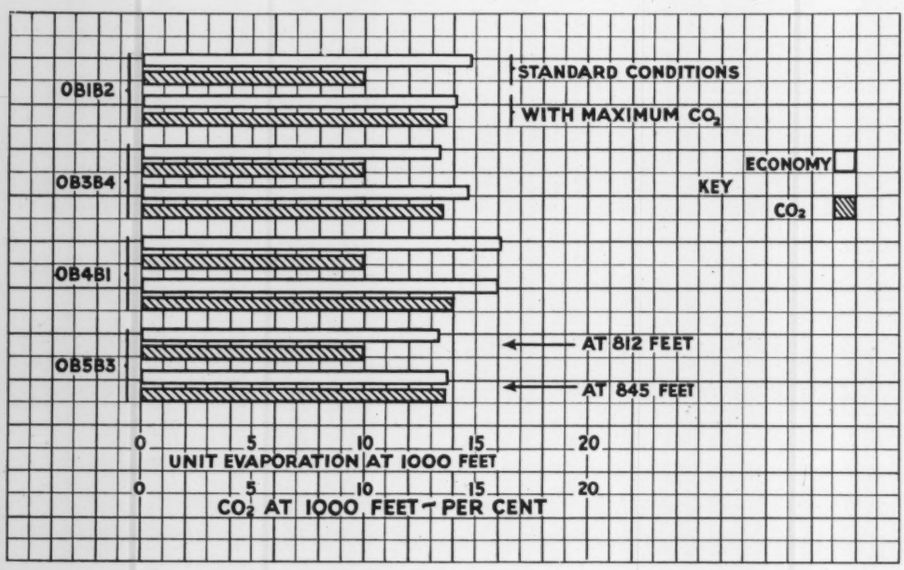


FIG. 4A—ECONOMY AND PER CENT CO_2 WITH HIGH DRAFT (SERIES *C*)

the burner. By the term furnace is meant the space in which the fuel is burned and corresponds to the term firepot as used with solid-fuel burning boilers. If the desired rate of fuel consumption cannot be obtained with a draft of 0.02 in. of water it is increased just enough to get the required kind of combus-



falls below 10 per cent CO_2 in all cases, and that the economy of the boiler is not as good. The fact that the decrease is not the same for all cases may represent a burner characteristic, but definite proof is lacking.

Series *D* represents only one test

FIG. 5A—ECONOMY AND PER CENT CO_2 AT MAXIMUM CO_2 (SERIES *D*)

run at an output of about 1000 ft. of radiation to demonstrate the best that the oil burner can do. The combustion is improved as much above 10 per cent of CO_2 as is possible without smoke or CO and the economy determined. This provides an actual performance of the

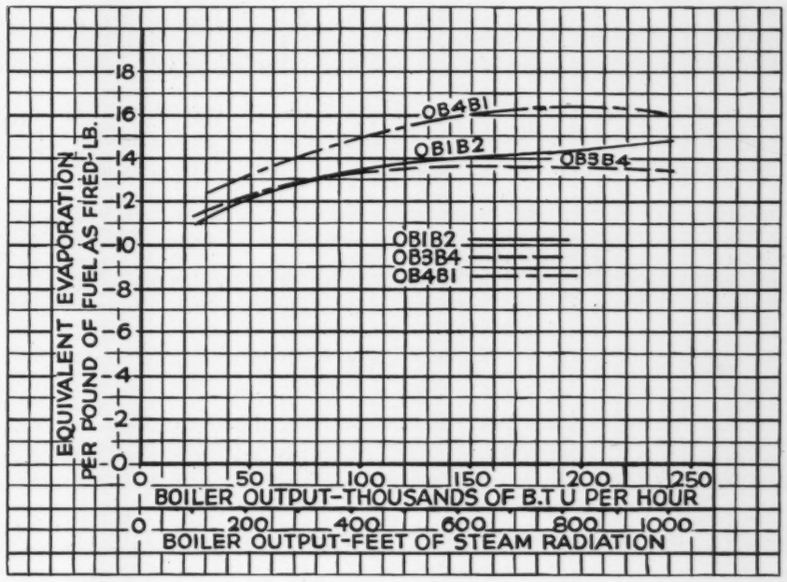


FIG. 6A—ECONOMY COMPARISON OF INTERMITTENT TESTS FOR STANDARD CONDITIONS (SERIES *E*)

highest order and offsets criticism of a 10 per cent CO_2 limit from manufacturers claiming superior performance for their burners. Fig. 5*A* shows the economy and CO_2 of the standard Series *A* and that obtained by Series *D*. From this figure it may be concluded that it is possible to have good combustion above 10 per cent of CO_2, but the gain is variable.

Series *E* represents a set of tests where the average boiler capacity is varied by intermittent operation of the burner. The burner is adjusted for a load of 1000 ft. of radiation under standard conditions of Series *A*. It is run for 30 minutes after conditions become uniform and then shut off various lengths of time to give varying boiler outputs. A 30-minute *on* period was chosen because it was found necessary to run that length of time to get the boiler and burner back to normal conditions of continuous operation. If this is not done, the conditions at the beginning and end of the test will not be the same, and the results will be in doubt.

This series is undoubtedly one of the most important because the burner and boiler will actually operate in this manner except with continuous type burners. It affords a reasonable picture of actual cellar operation.

Economy curves are shown in Fig. 6*A*. Only three curves are shown because one of the four burners tested to date is of the continuous type. The leading conclusion is that the average economy is reduced below that obtained in Series *A*. Again there is not sufficient information to date to determine how much of the decrease is due to the burner characteristics and how much due to the boiler.

Series *F* is a set of two tests run with intermittent operation similar in general character to Series *E* except that a furnace draft of 0.1 in. of water is used. This is intended to show the effect of high furnace draft with intermittent operation. It is quite obvious that a long *off* period with a high furnace draft will give the poorest economy results that can be obtained. Just as Series *D* is designed to show the best economy, Series *F* will show the worst. This procedure is not only fair, but is quite necessary in order to indicate how important the factors affecting operation are and which ones are most pertinent in achieving the best results.

Fig. 7*A* shows a set of three economy curves obtained by Series *F*. The obvious conclusion is that a higher furnace draft reduces the economy below that obtained by Series *E*. Again more sets of tests will be required before comparisons of various burners and boilers can be made.

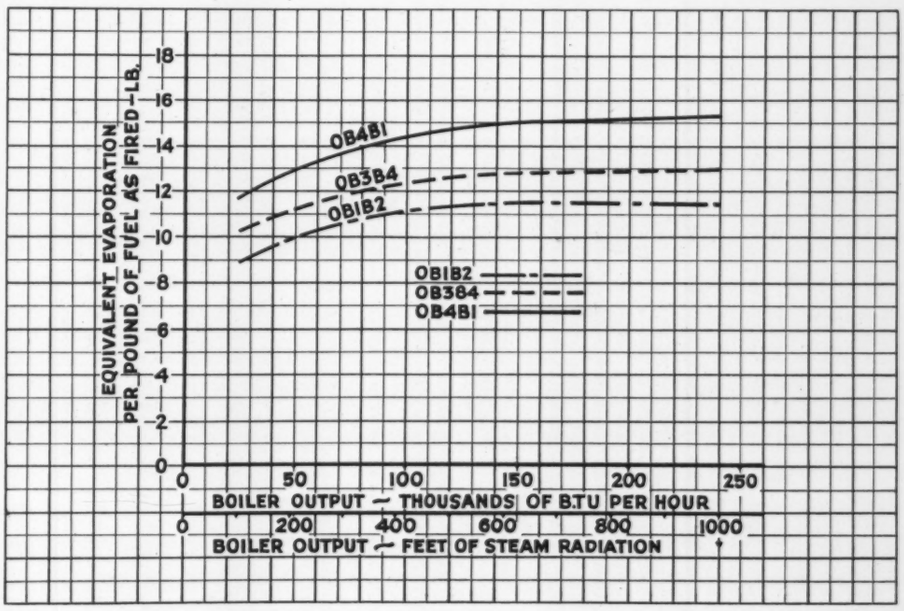


FIG. 7A—ECONOMY COMPARISON OF INTERMITTENT TESTS FOR 0.07 IN. DRAFT (SERIES *F*)

Series *G* represents a departure from the preceding tests in that it is designed to measure sound with and without combustion. The burner is set for a 1000 ft. load at the standard conditions of Series *A* and is run without combustion to determine air and mechanical sounds. Then with combustion occurring another reading is taken to measure air, mechanical and combustion sounds. A device loaned by the Bell Laboratories is being used to do this. The interesting point is that the sound units are the same as those being used to study the noises in New York City at the present time. This venture is a speculative one, but it promises to give comparative values. Fig. 8*A* shows the results obtained. An obvious conclusion is that the burners do not all produce the same degree of sound. The sounds are measured in terms of sensation units which have been standardized by the Bell Laboratories. Whether or not the same oil burner will produce measurably different sounds in different boilers cannot be answered yet. The power requirements of the burners are also measured and are shown graphically by Fig. 9*A*, which is self-explanatory.

The object of the foregoing discussion is to explain the kind of tests being made. They suggest the possibility of a number of valid comparisons that may be obtained after each burner has been tried in each boiler. At the present time there are four boilers and five burners available for tests. This calls

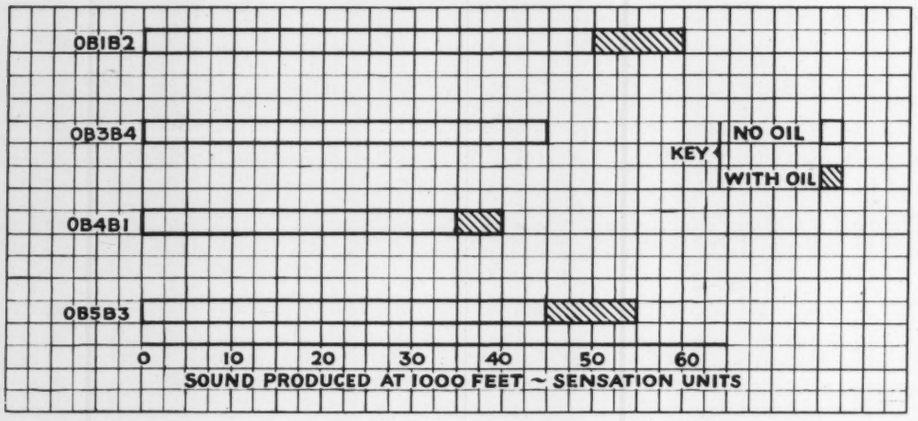


FIG. 8A—SOUND PRODUCED BY THE BURNER WITH AND WITHOUT COMBUSTION AS MEASURED BY THE AUDIOMETER (SERIES *G*)

for twenty complete series of tests of the kind just described. Four of the twenty are nearly complete. These four are therefore only representatives and only tentative conclusions should be made at this time.

The testing is going on steadily and it is hoped that the twenty sets will be finished in time for a more complete report at the Semi-Annual meeting of the A. S. H. V. E. in Minneapolis, June 24-27, 1930.

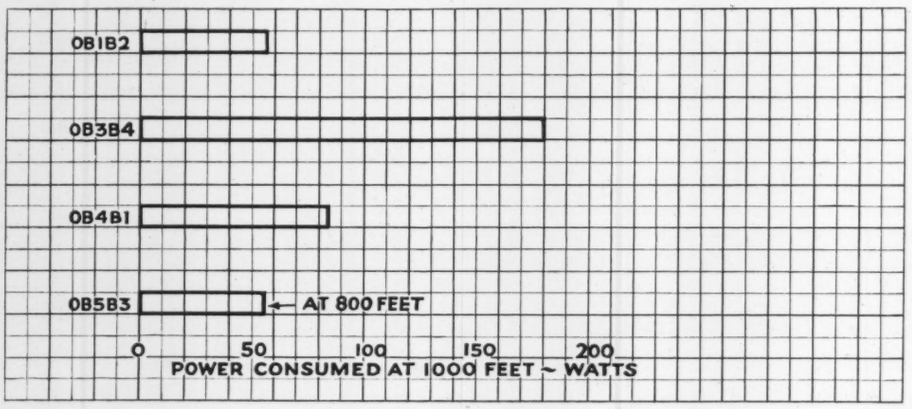


FIG. 9A—POWER CONSUMPTION FOR CONTINUOUS OPERATION

Boiler Rating Code Adopted as Society Standard

In accordance with the action taken by the members of the Society at the 36th Annual Meeting in Philadelphia, the Secretary sent to each member of the Society, a ballot, in order that he might vote *for* or *against* the Report of the Continuing Committee on the Code for Rating Steam Heating Solid Fuel Boilers.

The polls closed on April 1, and in advance of that date President Harding appointed a Board of Tellers to count the votes, consisting of W. H. Driscoll, *Chairman,* D. H. Faulkner and Raymond Newcomb, all of New York.

The report of the tellers is as follows:

TO THE COUNCIL
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
Gentlemen:

Your Board of Tellers appointed to count the ballots cast for the Code on Rating Low Pressure Steam Heating Solid Fuel Boilers are glad to say that the total vote cast numbered 700.

In accordance with the regulations governing this ballot, 499 legal ballots were found, 438 in favor of the adoption of the Code and 61 against adoption.

Respectfully submitted,
W. H. Driscoll, *Chairman*
D. H. Faulkner
Raymond Newcomb

April 2, 1930

Papers Presented Before the A.S.H.V.E. Classified According to Subjects

The following papers, on the subjects indicated, have been presented before the A. S. H. V. E., from the years of 1895 to 1929, inclusive. The volume number and year of the TRANSACTIONS in which each paper has been published are given in parentheses. Written and oral discussions entered into at regular meetings of the Society, which often are equally as important as the paper, also appear in the TRANSACTIONS immediately following the paper.

Reprints at present available are indicated by an asterisk (*), but these reprints do not always include the discussions. A complete list of papers, arranged according to the volumes of the TRANSACTIONS in which they appeared, was published in the April 1930 JOURNAL of the A. S. H. V. E. With the exception of Vol. 2, 1896, the TRANSACTIONS are available for every year to and including 1928. The 1929 TRANSACTIONS will be available in bound form at an early date.

Air Cleaners and Washers, Dust and Dust Removal

Removal of Dust from Boiler Rooms, by A. A. Cary (Vol. 6—1900)
*Dust in Relation to Heating, by Konrad Meier (Vol. 18—1912)
Methods of Automatic Humidity Control for Air Washers, by J. I. Lyle (Vol. 18—1912)
*Preliminary Report of Committee for Standardizing a Method of Testing Air Washers, by A. E. Stacey, Jr. (Vol. 20—1914)
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Proposed Method for Comparison of Effectiveness of Indirect Heating Surfaces, by A. E. Stacey, Jr., and C. M. Ashley (Vol. 31—1925)
Railway Passenger Car Heating, by E. W. Rietz (Vol. 23—1917)
Reasons for Failures of Heating Systems, by J. D. Hoffman (Vol. 24—1918)
*Reasons Why the Science of Heating and Ventilation Should be Observed, by J. D. Hoffman (Vol. 14—1908)
Recirculation of Air in Ventilation, by E. S. Hallett (Vol. 29—1923)
Relation between the Architect and the Heating Engineer, by T. N. Thomson (Vol. 5—1899)
*Relative Humidity, Its Effect on Comfort and Health, by J. I. Lyle (Vol. 18—1912)
*Re-Modeling of a Heating System, by Helen R. Innis (Vol. 25—1919)
*Report of Special Committee on Blower Systems for Heating and Ventilating, by A. M. Feldman (Vol. 20—1914)
*Report of Work of Ventilation Division of Chicago Health Department, by E. V. Hill (Vol. 20—1914)
Report of the Warming and Ventilation of Schools in England, by D. M. Nesbit (Vol. 5—1899)
*Review of Some Heating Literature, by H. J. Barron (Vol. 10—1904)
Safeguarding our Profession, by W. M. Mackay (Vol. 14—1908)
Save Coal in the Home, by L. P. Breckenridge and E. H. Lockwood (Vol. 24—1918)
*Selecting and Operating Laundry, Kitchen and Hospital Equipment, by H. C. Russell (Vol. 35—1929)
Separation of Oil and Grease from Exhaust Steam, by W. J. Baldwin (Vol. 3—1897)
Ship Ventilation, by F. R. Still (Vol. 26—1920)
Short History of our Society, by S. A. Jellett (Vol. 10—1904)
*Simplifying the Installation and Operation of School Heating and Ventilation Apparatus, by S. R. Lewis (Vol. 14—1908)
Smoke Prevention, by P. P. Bird (Vol. 17—1911)
Society, Its Aims and Opportunities, The, by J. J. Blackmore (Vol. 20—1914)
Some Comments on Present Day Heating and Ventilation Practice, by W. S. Timmis (Vol. 30—1924)
Some Points Regarding the Ventilation and Heating of Tall Buildings, by H. C. Meyer, Jr. (Vol. 5—1899)
Spontaneous Combustion, by M. W. Franklin (Vol. 24—1918)
*Street-Car Ventilation, by W. Thorn (Vol. 17—1911)
Temporary Warming of the Large Hall of the Royal and Imperial Palace Library in Vienna, by Edward Meter (Vol. 8—1902)
Test of the Beery System of Heating and Ventilating, by C. E. Beery (Vol. 26—1920)
Test of the Heating and Ventilating Plant, New York State Veterinary College, Cornell University, by R. C. Carpenter (Vol. 4—1898)
Theory and Practice of Heating and Ventilating in France, by G. Debesson (Vol. 20—1914)
*Time Analysis in Starting Heating Apparatus, by R. C. Taggart (Vol. 19—1913)
*Time Lag as a Factor in Heating Engineering Practice, by James Govan (Vol. 35—1929)
*Time Element in Heating Apparatus, by J. A. Donnelly (Vol. 18—1912)
Time Limit and Dry Walls Necessary in Testing in Heating Plant, by John Gormly (Vol. 8—1902)
Training of Janitors and Custodians, by E. S. Hallett (Vol. 26—1920)
Turbulence and Heat Transfer, by L'Roche G. Bousquet (Vol. 33—1927)
*Utility Demands of a Modern Hospital, by G. L. Larson, D. W. Nelson and R. A. Rose (Vol. 34—1928)
Utilization of Automatic Sprinkler System for Hot-Water Heating, by A. W. Moulder (Vol. 25—1919)
*Variability of Interior Conditions in School Rooms, by H. W. Whitten (Vol. 16—1910)
*Ventilation Division, Chicago Board of Health, by E. V. Hill (Vol. 19, 1913)
*Ventilation of the Debating Rooms of the New Riksdag's Building in Stockholm, by Wilhelm Dalgren (Vol. 12—1906)
*Ventilation of the Macy Store, New York, by D. M. Quay (Vol. 17—1911)
*Ventilation of Sleeping Cars—Comparative Tests of Various Types of Exhaust Ventilators, by T. R. Crowder, M. D. (Vol. 21—1915)
Ventilation Studies in Metal Mines, by D. Harrington (Vol. 27—1921)
What We Do and Don't Know about Heating, by J. R. Allen (Vol. 24—1918)
Windows, by E. V. Hill (Vol. 23—1917)

Tests of Secondary Air-Mixing Devices for Domestic Furnaces

The practical savings of fuel in house-heating and gains in smoke prevention resulting from the use of some of the air-mixing devices sold to householders to replace the ordinary furnace-door slots as supplied on boilers and furnaces are being determined by the United States Bureau of Mines, Department of Commerce.

Results of tests obtained by burning coke, anthracite and bituminous coal in a small steam heating boiler using natural draft show that the relative value of the device as compared with slots depends on the fuel condition, and might be an improvement or the reverse, depending on what quantity of secondary air was required; the devices tested so increased the resistance to the flow of air that the amount supplied was often insufficient. Tests were therefore made to determine the relative resistance and the quantity of air passing through the various devices and slots as related to the draft in the fire-pot. Wide differences were found; the largest of the devices only admitted half the air possible with the door slots wide open.

Further tests will be made to determine the results where the quantity of air required for complete combustion is within the capacity of the device.

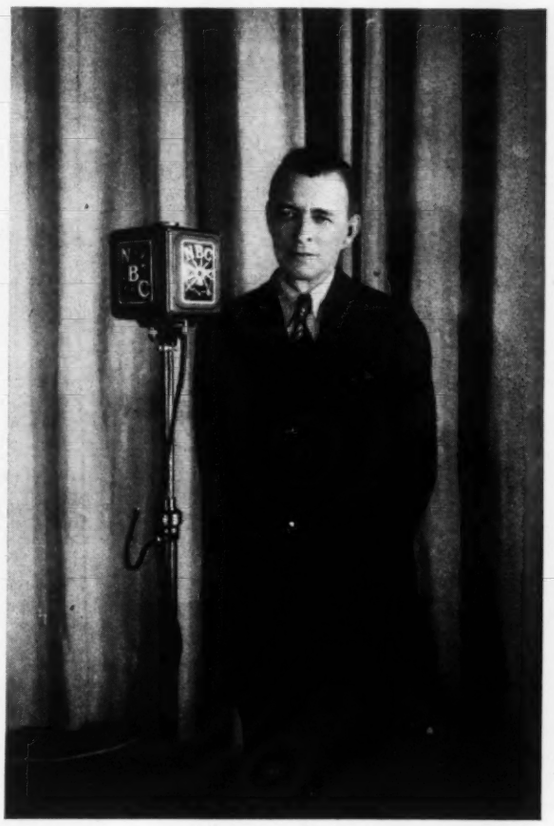
Heating and Ventilating Congress to be Held in Brussels

A Heating and Ventilating Congress will be held in Brussels, Belgium, June 26-29, inclusive, and an invitation has been extended to the members of the Society to attend this meeting.

It is hoped that as many of the American members of the Society as possible will attend. It is unfortunate that the dates of this meeting conflict with the dates of the Semi-Annual Meeting of the Society in Minneapolis, June 24-27, inclusive.

It is planned to have a special dinner for all the European and American members of the Society. Arrangements are in charge of Mr. Auguste Beaurrienne, 19, Rue Blanche (IXe), Paris, France, and it is suggested that all members of the Society who are planning to attend this Congress communicate direct with Mr. Beaurrienne.

PRESIDENT'S PAGE



PRESIDENT L. A. HARDING
AT THE MICROPHONE
Station WJZ, New York, April 1, 1930

THE HEATING AND VENTILATING INDUSTRY

An address broadcast over a chain of thirty-one stations of the National Broadcasting Corp.

IT IS my pleasure, through the courtesy of the Westinghouse Electric and Mfg. Co., to be permitted to briefly speak for a great industry, which contributes so much to our comfort, health and happiness.

The first effective co-operative endeavor made to remove the heating and ventilating industry, as a whole, from uncertain engineering practice occurred somewhat over a third of a century ago. A small group of nationally-known engineers and technical educators founded the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the express purpose of improving this art through the interchange of ideas, and to stimulate scientific research and invention. This society now numbers its membership in thousands, including engineers, physicians, educators, scientists and business leaders.

Certain combinations of various mechanical and, frequently, electrical devices that are employed to produce a comfortable temperature, and healthful atmospheric condition is termed a heating or ventilating system or both, as the case may be. The United States has, for many years, maintained the world leadership in this field of industry and today we manufacture more equipment for this purpose than all other countries combined.

It has only been through a correct interpretation and understanding of various physical laws, as developed by research, that we are able today to design and construct heating and ventilating systems which will produce predicted results. The following list are but a *few* of the outstanding research achievements in this field of engineering endeavor:

The determination of the heat transfer of building materials, building insulation and radiators.

The physical law governing psychrometric changes in the atmosphere.

Deodorization by the employment of ozone.

Physical laws governing the flow of water, steam and air in pipes and ducts.

The Hill synthetic air chart as a measure of ventilation, and the development of the so-called comfort chart, the measure of comfort as determined by temperature, humidity and air motion.

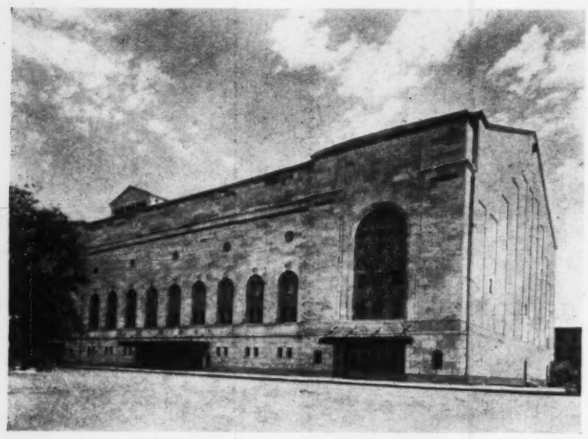
It is through the knowledge gained by research of this character that engineers are able to design apparatus, through the medium of which it is possible to control the heat and moisture loss from the body and thereby producing the state of well being or quiet enjoyment called comfort.

It has made possible the satisfactory heating and ventilation of our schools, theaters, the halls of our national legislative bodies, the Holland tunnel and the operation of deep mines. It is responsible for the improvement made in the quality of many manufactured products and the elimination of the economic loss due to material waste and employes time occasioned by uncontrolled temperature and humidity.

The day is apparently not far distant when all structures designed to house human beings will be equipped with automatically controlled heating, ventilating and air conditioning systems, which will produce a comfortable and healthful atmosphere the year around regardless of the outside temperature and relative humidity. No other building equipment can be responsible for an equal degree of satisfaction or dissatisfaction to the occupant, and it should, therefore, receive the greatest consideration when planning a new building of any character.

It remains for future scientific research to obtain for us a more thorough understanding of the vital characteristics of the air in their relation to health and disease, as supplied by nature and as may be artificially created. This is practically a virgin field for research and, if I may be permitted to make a prediction, will be responsible for some remarkable discoveries which will be applied for the benefit of mankind.

Minnesota Chapter Awaits You in June A.S.H.V.E. Semi-Annual Meeting



THE NEW MINNEAPOLIS AUDITORIUM

MINNEAPOLIS will be the scene of the Society's Semi-Annual Meeting, June 24-27, 1930, and headquarters for the meeting will be at the Curtis Hotel at Tenth Street and Fourth Avenue, South. Society members who attended the Annual Meeting in Philadelphia will recall the cordial invitation from the Minnesota Chapter, delivered by Albert Buenger, who is Chairman of the Committee on Arrangements and will direct the various activities which have been planned by the Minnesota members.

Membership of the Minnesota Chapter is drawn from the Twin Cities, Minneapolis and St. Paul, which form the Gateway to Minnesota's Ten Thousand Lakes. Within the city limits of Minneapolis are six lakes and the city has many beautiful parks connected by tree-bordered boulevards, whose charm and beauty unfailingly impress visitors to the city. The surrounding country is famous in history and legend, particularly in Indian warfare and the story of Hiawatha.

In addition to the beauty of the surrounding country, Minneapolis is a large manufacturing city and is the world's chief flour producer, maintains the leadership in the production of linseed products and has many manufacturing plants making equipment used by the heating and ventilating profession.

The University of Minnesota is located on the east bank of the Mississippi River, covers an area of 108 acres and houses more than 8,000 students in its 30 buildings.

A delightful program is being prepared for the membership who will attend the Semi-Annual Meeting, as the Minnesota Chapter members have unsurpassed natural beauties to show their guests and unlimited facilities for recreation.

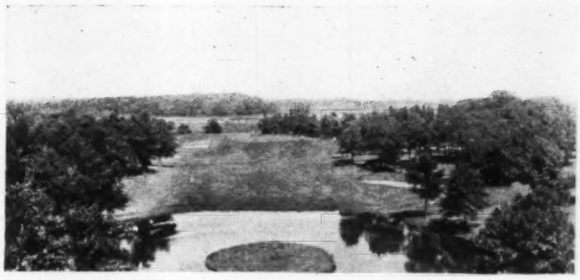
The comfort of the visitors to Minnesota this summer will be in the hands of the General Arrangements Committee, consisting of Albert Buenger, Chairman, H. E. Gerrish, A. J. Huch, F. B. Rowley, H. J. Sperzel, W. F. Uhl and A. M. Wagner.

Mr. Buenger is a mechanical engineer for the architectural firm of C. H. Johnson, is president of St. Paul Engineering Society and has been president, treasurer and secretary of the Minnesota Chapter. He also has served on many of its local committees as well as on Society committees.

Prof. F. B. Rowley, who heads the Reception and Registration Committee, is well known to Society members as Second Vice-President, Chairman of the Committee on Research and for his many contributions to the technical activities of the organization.

BY THE WATERS OF MINNETONKA

SAIL-BOATING ON LAKE CALHOUN,



GOLF IS POPULAR.

Hotel and transportation arrangements will be handled under the direction of W. F. Uhl, who is an active member of the Minnesota Chapter and is in the engineering supply business as a regular every-day activity.

The entertainment of the guests has been put in charge of Harry E. Gerrish, whose reputation is well established in Minneapolis for affairs that appeal to the tastes of the members and their guests.

Golf, that almost universal game, will be directed by Henry J. Sperzel, whose ordinary efforts are devoted to boilers, but who finds much enjoyment in arranging tournaments.

Publicity, which is related to selling, will be directed by A. J. Huch and the interesting stories which are forthcoming will emanate from his committee.

Finances, which are a necessary part of any meeting, will be handled by A. M. Wagner and his committee, whose chief duty as watchdogs of the treasury will be to see that everyone gets his money's worth.

The hostesses for the ladies will be directed by Mrs. F. B. Rowley as Chairman of the Committee and it is hoped that a large number will be in attendance.

In the customary manner for Summer Meetings, one session a day will be held and the technical program is developing under the guidance of the Program Committee, composed of A. C. Willard, Chairman, R. H. Carpenter and C. F. Eveleth.

Subjects for discussion will be varied. One of the papers already printed is Wall Surface Temperatures, by A. C. Willard and A. P. Kratz, and among those promised are Air Conditioning—A Vital Problem in Modern Bakeries, by W. L. Fleisher; Oil Burner Research, by L. E. Seeley; Control Equipment for Gas Burning Appliances, by W. E. Stark; and Research Papers from the laboratories of the University of Wisconsin and the University of Minnesota.

The way to Minneapolis is clearly marked and those who travel by train will find reduced rates attractive and for those who decide to motor, good roads converge from all points. Arrangements have been made to travel via the Chicago, Milwaukee, St. Paul and Pacific Railroad from Chicago to Minneapolis, and a special train will leave the Union Station, Chicago, at 7:30 p. m., June 23. The illustrations shown were furnished through the courtesy of this railroad.

For members, who can make the last week in June a vacation time, Minnesota offers many attractive tours and whether or not you have attended a summer meeting, remember there is both instruction and entertainment in making the trip.

PRELIMINARY PROGRAM

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
JUNE 24-27, 1930

Monday— A.M.—Headquarters, Reception Committee—LaSalle Hotel, Chicago, Illinois.
P.M.—Reception of Guests by Committee—Curtis Hotel, Minneapolis, Minn.

Tuesday— A.M.—Curtis Hotel.
8:30—Registration—Lobby floor.
9:30—12:30—Technical Session—Ball Room, Curtis Hotel.
11:30—Picnic Luncheon—Lake Harriet.
P.M.
2:00—Sightseeing and Inspection (Twin Cities and U. of Minn., Engr. Lab., Plant of Minneapolis—Honeywell.)
7:00—Council Dinner and Meeting.
8:30—Informal Reception and Dance, Curtis Hotel Ball Room.

Wednesday—A.M.
9:30—12:30—Technical Session—Ball Room, Curtis Hotel.
10:30—Ladies' Shopping Tour.
P.M.
2:00—Golf Tournament—Minneapolis Automobile Club. (Research Cup)
Ladies' Golf Match—Automobile Club.
Bridge and Tea for Ladies—Automobile Club.
7:00—Informal Dinner for Members and Ladies.

Thursday— A.M.
9:30—12:30—Technical Session—Ball Room, Curtis Hotel.
P.M.
1:30—Garden Party for Ladies.
2:00—Golf Meet (Kicker's Handicap) Thorp Country Club.
7:00—Semi-annual banquet and dance, Curtis Hotel Ball Room.

Friday— A.M.
9:30—12:30—Technical Session.
12:30—Hostess Luncheon for Ladies.

COMMITTEES FOR SEMI-ANNUAL MEETING 1930

General Arrangements Committee

A. Buenger, *Chairman*

H. E. Gerrish — H. J. Sperzel
A. J. Huch — W. F. Uhl
F. B. Rowley — A. M. Wagner

Reception and Registration Committee

F. B. Rowley, *Chairman*

N. D. Adams — C. E. Hasey
W. B. Clarkson — L. C. Hanson
S. A. Challman — E. F. Jones
G. A. Dahlstrom — J. V. Martenis
C. Foster — A. H. Probst
E. B. Gordon, Jr. — A. L. Sanford
M. S. Wunderlich

Hotel and Transportation Committee

W. F. Uhl, *Chairman*

J. H. Brown — H. G. Helstrom
E. J. Burns — W. N. Parks
M. H. Bjerken — F. C. Winterer

Entertainment Committee

H. E. Gerrish, *Chairman*

H. H. Bradford — A. J. Huch
C. G. Burritt — R. B. Mosher

Finance Committee

A. M. Wagner, *Chairman*

E. F. Jones — G. C. Morgan
R. W. Otto

Publicity Committee

A. J. Huch, *Chairman*

C. E. Gausman — Fred Shernbeck
C. E. Hill — E. J. Uhl

Golf Committee

H. J. Sperzel, *Chairman*

D. M. Forfar — J. B. Harris
D. C. Ruff

Ladies and Hostesses Committee

Mrs. F. B. Rowley, *Chairman*

Westinghouse Orchestra and Chorus Broadcasting From WJZ, New York, April 1, 1930

President Harding Speaks on Westinghouse Salute Radio Program to Heating and Ventilating Industry

OVER the coast to coast network of the National Broadcasting Co., the Westinghouse Electric and Mfg. Co. broadcast a program directed to the Heating and Ventilating industry on Tuesday, April 1.

The highlight of this half-hour program was a talk by Pres. L. A. Harding, of the Society, who spoke from WJZ in New York.

This was one of the series of Westinghouse salutes to modern industry and the program opened with an original musical composition by the Westinghouse orchestra of 40 pieces, directed by Cesare Sodero. Between the musical and vocal numbers by a chorus of 20 trained singers, the Voice of the Interludes gave a vivid story of the heating and ventilating industry and its development.

In these brief sketches, Phillips Carlon described the transition from early methods of heating to the gigantic installations which provide heating and ventilating for modern buildings and industries. Attention was directed to the great modern ventilating installation which keeps the proper atmospheric condition in the Holland Tunnel connecting New York with New Jersey.

President Harding told of the founding of the Society, 35 years ago, to stimulate scientific research in heating and ventilating, and gave a few of the outstanding research achievements contributed by the Society since its organization. He pointed out that no other building equipment is responsible for an equal degree of comfort and satisfaction to the occupant than the installation of heating and ventilating systems, and predicted that future research would establish thorough understanding of the vital characteristics of air in relation to comfort and health. (Full text of Mr. Harding's talk appears on page 439.)

By special invitation a group of Council members and others in New York and vicinity were interested spectators.

Each week the Westinghouse Co. salutes an important American industry and future programs will pay tribute to the Chemical Engineers, Central Station Engineers, Iron and Steel, Railway and other industries.

Many members of the Society reported excellent reception of the heating and ventilating program and the stations which carry this program are as follows:

WJZ	New York	WHAS	Louisville
WBZ	Springfield	WSM	Nashville
WBZA	Boston	WMX	Memphis
WBAL	Baltimore	WSB	Atlanta
WHAM	Rochester	WAPI	Birmingham
KDKA	Pittsburgh	WSMB	New Orleans
WJR	Detroit	WKY	Oklahoma City
KYW	Chicago	KPRC	Houston
KWK	St. Louis	WOAI	San Antonio
WREN	Kansas City	KSL	Salt Lake City
WEBC	Duluth	KGO	San Francisco
WRVA	Richmond	KECA	Los Angeles
KOA	Denver	KGW	Portland
WBT	Charlotte	KOMO	Seattle
WJAX	Jacksonville	KHQ	Spokane
WIOD	Miami		

Bulletin CS12-29

Commercial standard specifications for domestic and industrial fuel oils have recently been issued by the U. S. Department of Commerce. This is Bulletin CS12-29. These specifications cover a light, medium and heavy domestic fuel oil, and light, medium and heavy industrial fuel oils for domestic and industrial oil burning equipment.

A. S. M. E. Fifty Years Old

CELEBRATION of the 50th anniversary of the *American Society of Mechanical Engineers* was held, April 5-9 in New York City, Hoboken, N. J., and Washington, D. C. Pres. L. A. Harding, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and First Vice-President W. H. Carrier were the official delegates and conveyed greetings from the Society. They attended the functions in Washington, where Herbert Hoover, President of the United States, was honored as an engineer and was presented with the first Hoover Gold Medal, founded "to commemorate the civic and humanitarian achievements of Herbert Hoover and to be awarded hereafter for distinguished public service by engineers." The presentation was made by D. S. Kimball, President of the Board of Award and Past President of the *A. S. M. E.* The ceremonies in Washington were presided over by W. F. Durand and the speakers were Charles Piez, R. A. Millikan and J. V. W. Reynders.

Four other awards were made: The *A. S. M. E.* Medal for Distinguished Service in Engineering and Science was presented to William Leroy Emmet, consulting engineer of the General Electric Co.; the Gantt Medal for Distinguished Achievement in Industrial Management to Fred J. Miller, industrial engineer; the Guggenheim Medal for Notable Achievement in Aeronautics to Orville Wright; and the Melville Medal for an original paper or thesis of exceptional merit was presented to Prof. Joseph Wickersham Roe, New York University.

Fiftieth Anniversary Medals of the *A. S. M. E.* were conferred upon sixteen men of foreign nations, who were the authors of monographs, as follows:

AUSTRIA—Hofrat Ing. L. Erhard, Director, Technisches Museum, Vienna, Austria. Presented by Edgar L. G. Prochnik, Minister to the United States.

BELGIUM—Staff of six eminent engineers represented by Monsieur le Baron Gaston de Bethune, Engineer, Graduate from Artillery and Engineers School, New York, N. Y. Presented by Vicomte de Lantsheere.

CANADA—Brigadier General C. H. Mitchell, Dean, Faculty of Applied Science, University of Toronto, Toronto, Ontario, Canada. Presented by Vincent Massey, Minister to the United States.

COLOMBIA, SOUTH AMERICA—Prof. Julio Garzon Nieto. Presented by Enrique Olaya, Minister to the United States.

CZECHOSLOVAKIA—Dr. Stan. Spacek, Ministry of Public Works, Prague, Czechoslovakia. Presented by Ferdinand Veverka, Minister to the United States.

FRANCE—Georges Claude, General Manager, Liquid Air Company in Paris, France. Presented by Paul Claudel, Ambassador to the United States.

GERMANY—Dr. Ing. Conrad Matschoss, Director, Verein deutscher Ingenieure, Berlin, Germany. Presented by Fredrich W. von Prittwitz und Gaffron, Ambassador to the United States.

GREAT BRITAIN—Loughman St. L. Pendred, Editor, "The Engineer," London, England. Presented by Sir Ronald Lindsay, Ambassador to the United States.

ITALY—Senator Luigi Luiggi, Rome, Italy. Presented by Giacomo de Martino, Ambassador to the United States.

JAPAN—Dr. Masawo Kamo, College of Engineering, Tokyo Imperial University, Tokyo, Japan. Presented by Katsuji de Buchi, Ambassador to the United States.

MEXICO—Senor Ing. Norberto Dominquez. Presented by Manuel C. Tellez, Ambassador to the United States.

NETHERLANDS—Prof. Ir. D. Dresden, President, Jaffa Engine Works, Utrecht, Holland. Presented by F. H. van Royen, Minister to the United States.

SCANDINAVIA—Vilhelm Nordstrom, M. E., Vice Manager, AB. de Lavals Angturbin, Stockholm, Sweden. Presented by W. Bostrom, Minister to the United States.

SOUTH AMERICA—Prof. Donato Gaminara, Montevideo, Uruguay, Member of the Pan-American Highway Commission and member of Faculty of the University of Montevideo, and Prof. Julio Garzon Nieto, Chief Engineer, Office of Longitudes, Ministry of Foreign Affairs of Colombia. Presented by F. Varela, Minister to the United States.

SWITZERLAND—Prof. Dr. Aurel Stodola, Professor of Heat Engineering at Federal Technical High School, Zurich. Presented by Marc Peter, Minister to the United States.

UNITED STATES—Dr. C. E. Grunsky, President, American Engineering Council, Past President of American Society of Civil Engineers, Consulting Engineer, San Francisco, Calif. Presented by Joseph P. Cotton, the Acting Secretary of State.

Founders Medals were presented at a luncheon in the Mayflower Hotel by Robert L. Daugherty and those honored were Ambrose Swasey, one of the six survivors of the founders of the Society and a past president, John W. Cloud, London, John S. Coon, Atlanta, Robert F. Grimshaw and Francis H. Richards, New York, and E. H. Robbins, Pittsfield, Mass.

A White House Reception was held on Wednesday, April 9, the concluding day of the 50th Anniversary ceremonies which were described as one of the most impressive functions ever held by the engineering profession.

"The American Society of Heating and Ventilating Engineers extends to the American Society of Mechanical Engineers greetings and felicitations on the occasion of their fiftieth anniversary.

We desire to express our appreciation and admiration for the efforts of the American Society of Mechanical Engineers in extending the borders of engineering and for sponsoring the ideals of the engineer.

May you continue to grow in strength and ever exert a beneficial and kindly influence on the affairs of mankind."

L. A. Harding, President
American Society of Heating and Ventilating Engineers

New York, N. Y.
April 5, 1930.

Local Chapter Reports

Illinois

March 10, 1930. The March meeting of the Illinois Chapter was attended by 137 members and guests, who were honored by having among their guests Prof. A. C. Willard and Prof. A. P. Kratz, of the University of Illinois, and also T. J. Duffield, Executive Secretary of the Committee on Research of the Society.

Following the reading of the minutes of the previous meeting, which were approved as read, resolutions concerning the deaths of Charles D. Allan and of H. B. McLelland were placed on the Chapter records, and appropriately conveyed to the families of the deceased members.

The Better Relations Commission was then reported upon and discussed at length by H. G. Thomas.

The subject of Professor Willard's talk was the possible fallacy of so called *breathing line* temperatures, and the effect of two types of cast iron steam radiators on air temperatures in room heating, determined from tests made during the past few years at the University of Illinois.

In concluding, Professor Willard emphasized the point that a heating system, which for a given 5 ft. line temperature also produces the highest 2 ft. 6 in. line temperature and a relatively low temperature near the ceiling, is a far better system from the standpoint of both economy and comfort.

After some additional discussion of the subject by Professor Kratz, Mr. Duffield and Mr. Hale, the meeting continued to the subject of the proposed revision of the January 1929 Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers. This subject was very fully discussed by Professor Kratz, H. M. Hart, E. A. May, John Howatt and others, and at the conclusion the consensus of opinion seemed to be that the proposed revision should be approved.

The meeting adjourned after an enthusiastic vote of thanks was extended to the speakers of the evening.

Kansas City

March 10, 1930. Twenty-seven members and guests gathered at the Ambassador Hotel for the regular monthly meeting of the Kansas City Chapter of the Society, March 10, 1930, which was presided over by President Carl Clegg.

Following the reports of the various committees, A. W. Archer presented an illustrated lecture on the Mechanical Equipment of Modern Bakeries, covering problems of mechanical equipment as installed for economical production, adequate heating, refrigeration and ventilation. Mr. Archer pointed out that developing closer cooperation between architects, mechanical engineers, and heating and ventilating engineers, would be of great advantage to both the owner and engineer.

The A. M. Byers Co., Pittsburgh, Pa., then presented a four-reel picture descriptive of the Aston process of the manufacture of wrought iron pipe, perfected by one of the Society's members.

New York

March 17, 1930. Dr. Harvey M. Davis, president of Stevens Institute of Technology, Hoboken, N. J., was speaker of the evening and gave a most enlightening and entertaining talk on the subject of air pollution. The title of Dr. Davis' discourse was "Ventilating New York," and his remarks dealt primarily with outside atmospheric conditions rather than that of ventilation of the interior of a building. The subject was divided into three divisions, namely, turbulence, convection and advection.

Dr. Davis drew attention to the danger from respiratory diseases due to air pollution in our densely populated cities, and cited some figures to show that apparently very little can be done to "ventilate" these cities, but that a great deal can be accomplished in the way of reducing the sources of air pollution, one of the most common of which is the smoke nuisance. He expressed the opinion that it is the duty of the members of the Society to design and install efficient heat appliances, and with the proper capacities to minimize the effect of inefficient combustion, which is an important contributor to the smoke nuisance. Dr. Davis pointed out that although the subject is more or less controversial, there seems to be considerable evidence that air pollution in our larger cities has an important bearing on the health and longevity of the inhabitants.

The attendance at the meeting was 65. President Ritchie called attention to a Bill which is to come up for consideration in the State Legislature at an early date, the purpose of which is to amend the Labor Law in Relation to the Installation and Inspection of Steam Power and Heating Plants. A committee consisting of G. B. Nichols, *Chairman,* W. H. Driscoll and R. A. Wolff was appointed to investigate this Bill and to look after the interests of the Society.

Western New York

March 3, 1930. The March meeting of the Western New York Chapter of the Society was held at the Hotel Buffalo where 75 members and guests took part in the dedication ceremonies of the new Grill Room.

The Design and Construction of Chimneys was the topic given by the speaker of the evening, L. J. Summerhays, Rochester, N. Y. Mr. Summerhays illustrated part of his address with moving pictures which were very interesting, some showing the effects on chimneys of different design and construction in the average windstorm.

After a discussion on Mr. Summerhays' talk and after all important matters of business were taken care of, the meeting was adjourned.

Pacific Northwest

March 21, 1930. The monthly meeting of the Pacific Northwest Chapter of the American Society of Heating and Ventilating Engineers was held at the Washington Hotel, March 21, 1930.

Following the dinner, Prof. G. S. Wilson of the University of Washington presented a paper on Fan Performance, which contained valuable information obtained by a series of investigations made by Professor Wilson and W. L. Dudley, vice-president and general manager of the Western Blower Co. E. L. Weber, President of the Chapter, joined in the discussion which followed.

Earl Jones, of the Seattle Gas Co., spoke on the application of gas to the heating problems of Seattle, which was followed by a moving picture film showing freight handling operations of the American-Hawaiian Steamship Co.

It was agreed that Mr. Trowbridge, also of the Seattle Gas Co., postpone his talk until some later date.

W. B. Beggs, C. F. Twist and W. L. Dudley were appointed by President Weber to serve on the Nominating Committee for 1930-31.

There being no further business to attend to, the meeting was adjourned.

Philadelphia

March 13, 1930. The regular monthly meeting of the Philadelphia Chapter of the Society was held at the Engineers Club, March 13, 1930.

Secretary Davidson read the minutes of the preceding meeting, which were approved as read, and which was followed by the reading of the Treasurer's report.

R. V. Frost was then called upon to explain the features of the proposed revision to the Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers.

Mr. Frost gave a very comprehensive and detailed review of the various features included in the code and in this proposed revision, and strongly urged all members to cast their vote in favor of the code revision calling for heating boilers, of this type, to be rated by actual performance as opposed to the dimensional method advocated by makers of steel boilers. Thornton Lewis, J. D. Cassell and R. C. Bolsinger also spoke in favor of the code revision, and urged all members to cast their ballot in favor of the performance rating method.

A moving picture was then presented by Mr. Lang of the American Brass Co., showing all stages involved in the production of commercial brass and copper from the mines to the finished product. At the conclusion of the picture, a rising vote of thanks was extended to Mr. Lang and his company for their courtesy and cooperation in bringing this interesting and instructive information before the Philadelphia Chapter.

St. Louis

April 3, 1930. The regular monthly gathering took place on April 3, 1930, when 41 members and guests met for dinner, which preceded the meeting, at the Roosevelt Hotel.

President White called the meeting to order, followed by the minutes of the preceding meeting, which were approved as read.

J. M. Foster, reporting for the Program Committee, announced that Dean F. Paul Anderson of the University of Kentucky, Lexington, was to be the guest of the St. Louis Chapter at the May meeting.

President White then instructed the Resolutions Committee to write a letter to the President of the Society, L. A. Harding, congratulating him on the able manner in which he presented the subject of heating and ventilating in the short time allotted to him in the Westinghouse Radio Program.

The meeting was then turned over to E. B. Langenberg, who called upon G. A. Helwig to read a paper on the application of warm-air heating in connection with the industries. Considerable discussion followed Mr. Helwig's talk which was very interesting.

Mr. Langenberg was next on the program, presenting a paper covering the application of air heating in the home, predicting that eventually the heating plant for the home would be a complete air conditioning unit at one central location from which the air would be distributed to the various rooms.

After several discussions the meeting was adjourned.

March 5, 1930. The St. Louis Chapter of the Society held its regular dinner meeting at the Roosevelt Hotel on March 5, with an attendance of 46 members and guests.

At the conclusion of the dinner, President White called the meeting to order, and the minutes of the previous meeting were approved as read.

F. J. McMorran, reporting for the Research Committee, told of the research work being conducted at Washington University, which was most interesting to all.

Secretary Sodemann then read a letter from Pres. L. A. Harding and also an abstract of the report of the Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers, which was followed by considerable discussion, resulting in E. B. Langenberg offering a motion that the St. Louis Chapter go on record as a chapter favoring the recommended revisions of the committee. The motion was seconded and passed unanimously.

President White then introduced the guest speaker of the evening, E. A. Jones of the L. J. Mueller Furnace Co., Milwaukee, who gave a very interesting and instructive talk on Gas Heating. This presentation also created considerable discussion and many questions were asked.

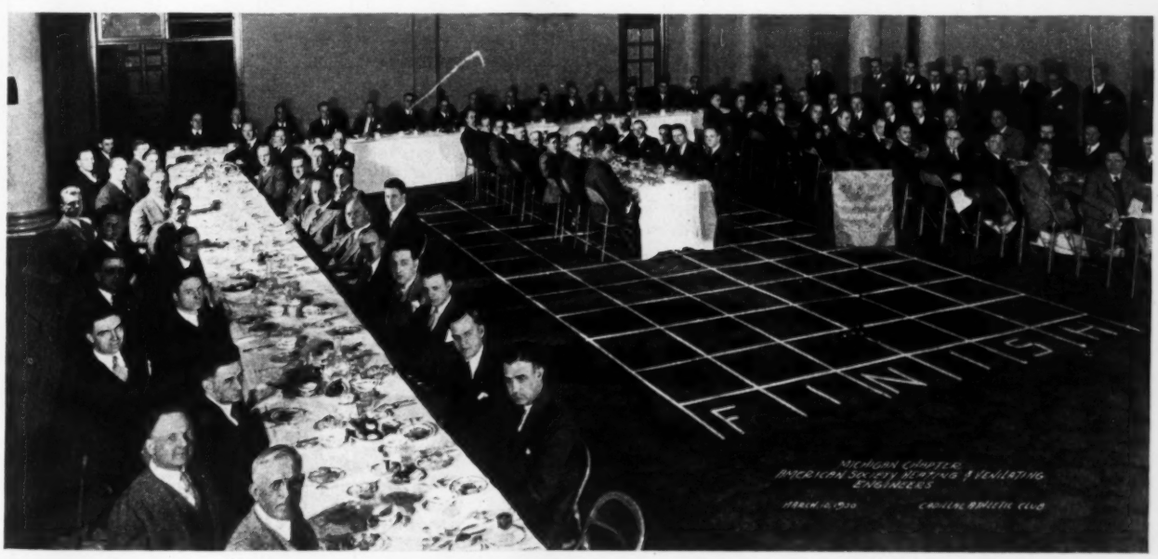
Following the discussion the meeting was adjourned.

Michigan

March 10, 1930. The Cadillac Athletic Club was the scene of the March meeting of the Michigan Chapter when 96 members and guests responded to the roll call.

Pres. William G. Boales requested all of the members who joined the organization during the last ten months to come forward to the speakers' table, where they were introduced and given a great hand by the assembly.

An explanation of the code for rating low pressure heating boilers was given by W. A. Rowe, a member of the Council,



FINISH OF THE DERBY COURSE

and additional comments were made by J. F. McIntire, E. M. Harrigan and N. B. Hubbard.

The speaker of the evening was Harvey Campbell, vice-president and secretary of the Detroit Board of Commerce, who predicted reasonable business activity for 1930 and a tremendous increase in business in 1931 and 1932.

At the conclusion of Mr. Campbell's speech, the meeting was placed in the hands of W. J. Whelan, chairman of the Program Committee, who staged the Annual A. S. H. V. E. Derby.

In the illustration on the preceding page, the finish of the Derby Course will be noted and in the foreground is the prize banner of the Michigan Chapter, won in the National Membership Campaign, 1929.

Wisconsin

March 23, 1930. The March meeting of the Wisconsin Chapter of the Society was held on Monday, the 23rd of the month, at the Milwaukee Elks Club, and was attended by 26 members and guests, with Pres. F. G. Weimer presiding.

In the absence of Prof. G. L. Larson, who was to be the first speaker of the evening, E. A. Jones, of the L. J. Mueller Furnace Co., Milwaukee, very ably discussed the subject of the proposed revision of the A. S. H. V. E. Boiler Code, telling the history of the Code from its beginning about five years ago, and continuing until its present form. He then told of the possibilities if the Code were to be adopted. Mr. Jones also stated that he believed that eventually it would come to the point where the A. S. H. V. E. Research Laboratory would test and rate all boilers similar to what the *American Gas Association* laboratory is doing with the gas-fired boilers. He stressed the point that the Code would prove most popular if the test were run by disinterested parties.

Before concluding his talk, Mr. Jones read the new steel heating boiler code, and explained that the steel boiler manufacturers were not in full accord with the proposed revisions and had asked that steel boilers be omitted from the Code.

The next speaker of the evening was C. V. Haynes, who spoke on the Society in general and referred to the success of the First Heating and Ventilating Exposition held in connection with the 36th Annual Meeting of the Society in Philadelphia in January.

Mr. Haynes then discussed in detail the history of THE GUIDE since its founding in 1922, mentioning that it is now used as a text book in many colleges and universities throughout the United States.

He then told of the new Society headquarters at 51 Madison Ave., in New York; he also suggested that each Chapter devote a meeting for discussion of THE GUIDE, and send in any suggestions for improving the book as the Guide Committee usually receives very little help from the outside.

Mr. Haynes touched upon the work done for improving the heating and ventilating industry through the work of the trade practice conference, and the boiler and radiator institute. He expressed the opinion that business for the year 1930 would be better than that in 1929, and made a plea for cooperation between manufacturers.

It was agreed by all that Mr. Haynes' talk was very inspiring and helpful, and a vote of thanks was extended to him, which was followed by adjournment.

New Corporation Announced

It has just been announced recently that Guy S. Faber has resigned from the Jas. P. Marsh Co., Chicago, and has formed his own company now being incorporated under the laws of the State of Illinois to be known as the General Regulators Corp., located at Kolmar and Grand Avenues, Chicago.

William Gage Snow Dies

The Society announces with sincere sorrow the death of one of its well known and well liked Past-Presidents, William Gage Snow, who died suddenly of heart failure at his home, 11 Devon Road, Newton, Mass., on the evening of February 14, 1930.

Mr. Snow was born in Watertown, Mass., February 15, 1866. He was graduated from the Massachusetts Institute of Technology in the class of 1888, after which he went to Philadelphia, where he devoted his time to the heating industry. After spending a few years in Philadelphia he returned to Boston, and was made the New England Manager of Warren-Webster & Co., which position he held for several years.

He joined the Society in December, 1903, and in 1909 became President. He devoted considerable time to the upbuilding of

WILLIAM G. SNOW

the Society and was one of our outstanding members, as well as one of the outstanding men of his time in engineering ability. One of his early contributions to the profession was a book on heating and ventilating which is in use today.

In April, 1918, he severed his connections with the heating business to become Treasurer of the Middlesex Products Co., manufacturers of fancy paper, of which his brother, Walter B. Snow, was president. Upon the death of his brother, he also became President, as well as Treasurer, filling both positions until his death in February.

Although he resigned from the Society in 1922, he maintained an active interest in its affairs and in the developments of the art of heating and ventilating, which have profited greatly by his contact.

The Council and members of the Society express their heartfelt sympathy to the widow, daughters and son, who survive him.

John H. Roberts Dies

It is with deep sorrow that we learn of the death on March 4, 1930, of John Hammond Roberts, one of the prominent members of the Society.

Mr. Roberts attended the New York Military Academy at Cornwall, N. Y., the Cleveland Engineering Institute, and also made an extensive study of heating from various text books on this subject.

For fifteen years he was associated in different ways with the heating industry, as a practical steamfitter, as a salesman of heating materials and as an engineer. At the time of his death he was with W. A. Case & Son Mfg. Co., Detroit, Mich.

Mr. Roberts had many friends in the Society as a whole and his loss is deeply regretted by his associates.

New Address

The Frank Irving Cooper Corp., Boston, Mass., formerly at 127 Tremont St., is now located in new quarters at 47 Winter St.

Council Meets at New Society Headquarters

The regular meeting of the Council was held April 2 at 51 Madison Ave. in the new headquarters office of the Society.

President Harding presided and those in attendance were as follows: D. S. Boyden, Boston; R. H. Carpenter, New York; W. H. Carrier, Newark; J. D. Cassell, Philadelphia; Roswell Farnham, Buffalo; W. T. Jones, Boston; Thornton Lewis, Philadelphia; F. C. McIntosh, Pittsburgh; W. A. Rowe, Detroit; F. B. Rowley, Minneapolis; and A. V. Hutchinson, Secretary.

Plans for Annual Meetings in 1931 and 1932 were discussed and invitations were received from the Pittsburgh and Cleveland Chapters.

One of the most important items on the program was consideration of the budget for 1930 and the recommendations of the Finance Committee were approved and a budget was adopted.

The necessity for several continuing committees was pointed out by President Harding and appointments are to be made as follows: Committee on Code for Testing and Rating Non-Ferrous and Concealed Radiators; Committee on Code for Testing and Rating Unit Ventilators; and Code of Minimum Requirements for the Heating and Ventilation of Buildings.

The resignations of seven members were accepted and seventy were dropped for non-payment of dues.

Panel System Representatives Announced

Wolff and Munier, Inc., 222 East 42nd St., New York City, have been appointed by Richard Crittall & Co., Ltd., of London, as licensees in the United States for the panel system of heating. This system provides radiant heat from ceilings or walls by means of concealed pipes using hot water as the heating medium. The Panel System was described in a paper entitled Panel Warming presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers in Philadelphia, January, 1930, by L. J. Fowler, London, England.

Past-Presidents

At each annual meeting, the Past Presidents of the Society have a formal meeting and at the 36th Annual Meeting in the Benjamin Franklin Hotel, Philadelphia, it was appropriate that they meet in the Benjamin Franklin Room and dine under the gaze of the Society's Patron Saint, Benjamin Franklin. The group attending this year is shown in the illustration at the bottom of this page.

Committee on Water Heating Organizes

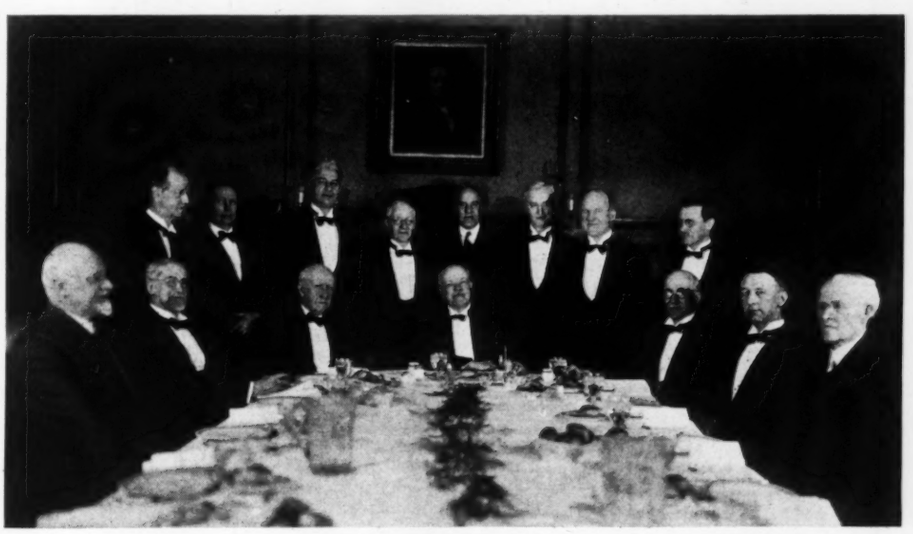
A two-day meeting, March 27 and 28, was held at the U. S. Bureau of Mines, Pittsburgh Experiment Station, on the subject of volume water heating for domestic purposes in houses, apartments, hotels, office buildings and hospitals.

This meeting, sponsored by the Committee on Volume Water Heating of the *American Gas Association*, had for its object the development of standard practice and methods of calculating hot water heating requirements based on the number of occupants, rooms, apartments or general building characteristics.

Those in attendance were: C. W. Berghorn and C. H. Light, *American Gas Association;* F. C. Houghten, A. S. H. V. E. Research Laboratory; E. J. Horton, Ruud Mfg. Co.; E. A. Jones, L. J. Mueller Furnace Co.; S. J. Lonergan, Bastian-Morley Co.; H. J. Long and Mr. Yost, Kompak Co.; M. S. Maehling, Equitable Gas Co.; H. P. Morehouse, Public Service Electric & Gas Co.; M. Resek, Perfection Stove Co.; and F. D. Tansey, Brooklyn Union Gas Co.

American Oil Burner Association Elect President

Walter F. Tant, Detroit, president of the Silent Automatic Corp., and a prominent figure in the development of the oil burner industry, became the seventh president of the *American Oil Burner Association* on April 10, when he was elected at the annual convention of the association in Chicago to succeed E. M. Fleischmann, of Baltimore. Mr. Fleischmann has served as president for the past twelve months with Mr. Tant serving as senior vice-president. The newly-elected president has had a continuous experience in the administrative affairs of the Association since April, 1926, when he was first elected to the board of directors. His election to the highest office in the gift of the industry is a recognition of his contributions to its advancement.



SEATED—Left to Right: Homer Addams 1924, R. P. Bolton 1911, S. A. Jellett 1895, F. Paul Anderson 1927, J. R. McColl 1922, H. P. Gant 1923, D. M. Quay 1900. *STANDING*—Left to Right: D. D. Kimball 1915, E. V. Hill 1920, J. I. Lyle 1917, J. F. Hale 1913, H. M. Hart 1916, S. R. Lewis 1914, W. H. Driscoll 1926, S. E. Dibble 1925

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 17 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 15, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Bailey, W. Mumford, Co-Partner, British Trane Co., London, England.	R. N. Trane T. R. Johnson	R. H. Anderegg M. W. Miller
Carson, Clifford Clyde, Skinner Bros. Mfg. Co., Inc., St. Louis, Mo.	W. A. Russell Henry Baetz	E. R. Barnes J. L. DeNeille
Cartier, Charles Ernest, Scott Stores, Inc., Chicago, Ill.	B. L. Casey H. J. Lagodinski	L. L. Narowetz, Jr. J. F. Hale
Chandler, Clark W., Carnegie Inst. Tech., Pittsburgh, Pa.	S. E. Dibble E. C. Evans	E. N. Pettit, Jr. K. E. Johnson
Green, Robert F., Kelsey Heating Co., New York, N. Y.	C. R. Place R. H. Bradley	O. O. Oaks E. A. Scott
Greene, Alfred Reynolds, Heating Contractor, Binghamton, N. Y.	W. F. Johnson Joseph Davis	H. B. Roarke J. J. Landers
Hayman, A. Eugene, Jr., Carnegie Inst. Tech., Pittsburgh, Pa.	S. E. Dibble E. C. Evans	E. N. Pettit, Jr. K. E. Johnson
Kuehnert, Albert C., Wichita Gas Co., Wichita, Kans.	J. H. Kitchen F. A. Kitchen	G. W. Smith J. C. Miles
Leach, Thomas F., Htg. Engr., Philadelphia Electric Co., Ardmore, Pa.	A. E. Kreibel M. F. Blankin	R. E. Jones F. W. Wandless
MacKenzie, Donald Waterman, Holland Furnace Co., Pittsburgh, Pa.	E. C. Evans V. W. Cherven	G. W. S. McEllroy F. A. Gunther
Norris, William Dauncey, The John R. Kehm Co., Chicago, Ill.	V. H. Walther R. B. Hayward	J. F. Hale S. R. Lewis
Olson, Gilbert E., Vice-Pres., Olson Bros., Omaha, Nebr.	R. E. Atkinson M. D. Pence	W. J. Mauer M. S. Good
Partridge, George Allen, Sales Engr., Herman Nelson Corp., Moline, Ill.	Phillips Eaton A. P. Goodhue	J. S. Webb W. T. Jones
Pelletier, Albert, S. R. Lewis, Chicago, Ill.	V. H. Walther S. I. Rottmayer	S. R. Lewis H. G. Kreissl
Sebree, George M., Carrier Engrg. Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	I. I. Lyle M. S. Smith
Summers, Ernest Trevor, Cotter Bros., Winnipeg, Manitoba, Canada.	W. W. Hicks H. H. Angus	H. J. Church J. S. Wood
Viessman, Warren, Bureau of Mechanical-Electrical Service, Baltimore, Md.	R. L. Leilich James Posey	C. L. Reeder Hermann Eisert

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ATHERTON, F. R., American Radiator Co., Detroit, Mich.

BRAATZ, CHESTER JOHNSON, Instr., Steam & Gas Engrg., Univ. of Wisconsin, Madison, Wis.

BROWN, JAMES LESTER, Partner, Eames & Brown, Pontiac, Mich.

BROWN, TOM, B. F. Reynolds & Co., Detroit, Mich.

CANTWELL, JOHN DEMUTH, Engr., Trane Co., and A. Harvey's Sons Mfg. Co., Detroit, Mich.

CONOVER, EDMOND W., Research Engr., Detroit Steel Products Co., Detroit, Mich.

COOK, RALPH P., Htg. & Vtg. Engr., Eastman Kodak Co., Rochester, N. Y.

COSSABOOM, GEORGE C., Weston & Ellington, Detroit, Mich.

EAMES, W. RUSSELL, Member of Firm, Eames & Brown, Pontiac, Mich.

HALLETT, SAMUEL G., Mgr., The Hallett Engrg. Co., St. Louis, Mo.

HARE, W. ALMON, Mgr., Stoker Div., Whitehead & Kales Co., River Rouge, Mich.

HARRISON, CHARLES GEORGE, Crane Co., Detroit, Mich.

HARTMANN, GEORGE F., Fallansbee Bros. Co., Pittsburgh, Pa.

HOSHALL, ROBERT HOUSTON, Associate of Thos. H. Allen, Cons. Engr., Memphis, Tenn.

INGLEFIELD, CHARLES L., Treas., Bushnell Mchy. Co., Pittsburgh, Pa.

KING, DAVID J., American Blower Corp., Detroit, Mich.

LANDERS, JOHN J., Sales Engr., U. S. Radiator Corp. & Pacific Steel Boiler Corp., Buffalo, N. Y. (*Advancement*)

LASALVIA, JAMES J., Chief Mech. Engr., Chas. S. Leopold, Philadelphia, Pa.

MCCLANAHAN, LUTHER C., Sales Engr., Buffalo Forge Co., Buffalo, N. Y.

MCCULLOUGH, JOHN LEWIS, Sales Engr., Direct Control Valve Co., Pittsburgh, Pa.

MEYER, CHARLES L., L. J. Wing Mfg. Co., New York, N. Y.

PARROCK, EDWIN J., American Blower Corp., Detroit, Mich.

PEACOCK, HERBERT, Dist. Mgr., Carrier Engrg. Corp., Detroit, Mich.

PITTOCK, LOUIS BURNS, 902 Oliver Bldg., Pittsburgh, Pa.

PURDY, ELLWOOD D., Pres., Hardinge-Detroit Corp., Detroit, Mich.

READER, JOSEPH THOMAS, Vice-Pres., Kerr Mchy. Corp., Detroit, Mich.

REED, VAN APPLEGATE, JR., Secy. & Treas., Federal Engrg. Co., Pittsburgh, Pa.

SANFORD, STERLING S., Htg. Engr., The Detroit Edison Co., Detroit, Mich.

SHAVER, ELMER W., Sales Engr., Coon-DeVisser Co., Detroit, Mich.

SHEA, MICHAEL B., American Radiator Co., Detroit, Mich. (*Reinstatement*)

TEELING, GEORGE A., N. Y. State Mgr., B. F. Sturtevant Co., Rochester, N. Y.

TETEREVNIKOFF, NIKOLAS, Prof., Polytechnic Engineering School, Leningrad, Russia.

TOWER, ELWOOD S., Chief, Engrg. Dept., American Radiator Co., Pittsburgh, Pa.

WARREN, WALTER JOSEPH, Smith, Hinchman & Grylls, Detroit, Mich.

ASSOCIATES

BYRNES, EMMET F., Mgr., H. Kelly & Co., Detroit, Mich.

CHITTENDEN, EDWIN FOSTER, Branch Manager, The Herman Nelson Corp., Moline, Ill.

CORNELL, J. CLARENCE, Draftsman, United Engineers & Constructors, Inc., Philadelphia, Pa.

COWAN, EDWARD, JR., The Cowan Co., Detroit, Mich.

DONOVAN, WILLIAM J., 2239 North 27th St., Philadelphia, Pa.

FISHER, ALBERT E., Sales Engr., Kerr Mchy. Corp., Detroit, Mich.

GALLARNO, CHARLES A., Secy., Treas., The Donald Miller Co., Detroit, Mich.

GIDDEY, YORK R. F., Vice-Pres. & Genl. Mgr., Bryan Boiler Sales, Inc., Detroit, Mich.

HASSBERGER, HENRY C., Lorne Plbg. & Htg. Co., Detroit, Mich.

HODGES, GEORGE S., JR., Detroit Stoker Co., Detroit, Mich.

JOHNSON, P. H., Muskegon, Mich. (*Reinstatement*)

LOUCKS, DAVID WENDELL, Duquesne Light Co., and Allegheny Steam Htg. Co., Pittsburgh, Pa.

MCNAMARA, WILLIAM, Sales Engr., The Trane Co., Minneapolis, Minn.

NENTWIG, RAY J., Murray W. Sales & Co., Detroit, Mich.

O'DAY, CORTLAND N., Mgr., Industrial Htg. Dept., Buckeye Blower Co., Columbus, Ohio.

POE, THOMAS O., Pres., Poe Co., Detroit, Mich.

RANDALL, ROBERT D., D. T. Randall & Co., Detroit, Mich.

REINKE, FLOYD C., American Insulating Co., Detroit, Mich.

SAUER, ROBERT L., Riley Stoker Corp., Detroit, Mich.

SKILLIN, N. MORTON, Dist. Sales Mgr., Cohoes Rolling Mill Co., Detroit, Mich.

SMITH, ROBERT CRAWFORD, Sales Repr., Columbia Radiator Co., Pittsburgh, Pa.

SWARTZ, HENRY A., Sales Engr., U. S. Radiator Corp., Detroit, Mich.

TRETHAWAY, RAY S., Sales Engr., Chicago Pump Co., Chicago, Ill.

JUNIORS

BUCHER, HARRY GEORGE, C. A. Dunham Co., Pittsburgh, Pa.

COUCH, NORMAN HAROLD, N. Y. Edison Co., New York, N. Y.

HEMPHILL, DAVID ANDREW, York Htg. & Vtg. Corp., Pittsburgh, Pa.

TELLER, WILBUR W., The Ric-WiL Co., Cleveland, Ohio.

Heating · Piping and Air Conditioning EDITORIALS

Waste and the Maintenance Engineer

For two decades or more the chemist, from an industrial and economic point of view, has been the hero of the hour. Now the maintenance engineer is receiving a nation-wide acclaim and by nomination of the President of the United States in 1930 is the key man in the movement for the maintenance of prosperity. In a recent address, President Hoover mentioned particularly the work of the maintenance engineer as one of the greatest forces for prosperity which our economic system affords.

It is obvious that industrial competition will increase rather than decrease. No longer will keen competition permit the maintenance engineer to ignore wastes that are common in many industrial plants. The result is that the most comprehensive elimination of waste program ever conceived of has been set in motion.

Among the chief concerns of a maintenance engineer are those that have to do with air conditioning, heating and piping. There are processed materials which may be affected by air conditioning so as to decrease waste or prevent their deterioration. Humidifying, dehumidifying, cooling, heating, control of temperatures and pressures have many undeveloped possibilities for conserving materials, for preventing decay or spoilage and for saving labor cost by providing employes with more healthful air environment. Industrial plants can not expect employes to produce as they should without adequate heating and air conditioning facilities. Partial illness, while working, or absence from work on account of respiratory diseases, can be corrected to great degree by proper air conditioning.

In line with the proposed program of elimination of waste, in which the maintenance man figures so prominently, here are some of the points which one might take into consideration: Since there is a difference of 70 per cent in the heating costs in factories comparable in size, it might be well to look into the efficiency of the heating plant. Is steam wasted which might be used for heating or for some other purpose? Is a suitable temperature maintained throughout the plant? Are there cold spots which could be made comfortable by unit heaters or other equipment? Are there objectionable drafts which cause discomfort or cold among the workmen? Is the ventilation automatic or is it left to the workman who thinks about it?

These are only a few out of many projects that might be undertaken by the maintenance engineer in order to tie in with this national movement for elimination of waste.

An Essential Part

The manager of the real estate department of a large corporation, owning numerous buildings throughout the United States, recently prepared ten useful rules for space renting. It is interesting to note that one of the guides he stressed is that "equipment means more than land or buildings."

If we consider the mechanical equipment of a structure as not a part of the building, but as an accessory, the rule is one of the truths we have all recognized but have not put into simple words. On the other hand, can we have a completed building without the proper heating equipment, the necessary piping, or (in these days of keen competition between large structures) the air conditioning system? Obviously not.

The heating, piping and air conditioning systems of a modern structure (industrial plant, office building, hospital, hotel or theater) are as much a part of the building as is the roof or the footings. We would revise the rule to say, "Without the proper mechanical equipment you have no building to rent."

Making Money

A couple of years ago we read a book on decreasing costs in industrial plants. Somewhere in the introduction, the author defined engineering as "the art of making money" and went on to point out that, if there is any distinction between saving money and making it, it is very slight.

It is a good idea to keep this definition in mind—especially should the engineer consider it when designing or installing a piping system. Very often, the waste caused by an incorrectly designed or installed system of piping is not apparent for some time. For example, in this issue of HEATING, PIPING AND AIR CONDITIONING, in an article on refrigeration piping, the author (a consulting engineer) tells of unreasonable pumping costs (because of incorrect piping) that were not discovered until the plant was re-arranged to drive the pumps electrically.

An investigation of the needless expenditure of money because of faulty piping should not be delayed until the day when other factors cause remodelling. Are your piping systems operating at the top of the efficiency curve now?

The Association for Correlating Thermal Research

"When business lags, the best way to stimulate sales is to devise something new and useful which will be welcomed by the public."

"Research is the cornerstone of industrial progress."

"The company which does not adopt the policy of research may as well set the date of its liquidation."

"I believe that research is the answer to the competition that is increasing every day."

"We find that research is the best weapon of offense and defense."

These are statements by officials of large business concerns. The truth of these statements is obvious to the more successful of them, but still is not accepted by many concerns who have not had the opportunity to observe the results of conducting scientific tests of material or equipment.

Many a concern has started a program of research without any thought of related research which already might have been conducted. There has been much duplication in research in universities. In many instances, inquiry has revealed, the head of one university research laboratory did not know what the other research laboratories were doing. There is much test data in the files of industrial concerns available to the engineer which never has been brought to light. There was no means for gathering together in one place all the heating, piping and air conditioning information on one subject, device or commodity. It was obvious, however, that before starting a program of research a concern should know what research along its particular line had already been conducted.

These conditions were brought to the attention of the *American Society of Heating and Ventilating Engineers* by Samuel R. Lewis, consulting engineer of Chicago at the 1927 summer meeting of the society. The result was the establishment of the *Association for Correlating Thermal Research,* which has its offices at the Bureau of Mines, 4800 Forbes Street, Pittsburgh, Pennsylvania. It is sponsored by several national organizations, including the *A. S. H. V. E., the Heating and Piping Contractors' National Association, American Gas Association,* and the *National Warm Heating Association.* The purpose of this association gradually is being extended and it recently published a booklet describing its activities. Among other information it includes a directory of university laboratories, private laboratories, government bureaus, technical societies, trade associations, and other organizations engaged in supporting or otherwise interested in research, codification, standardization or improvement of practice in the field of thermal engineering, with special reference to heating and ventilating and their relation to the health and comfort of man.

A small charge is made for this service which covers the cost of copying and incidental clerical work. An alphabetical file of four thousand articles on all phases of heating and ventilation, and their relation to human health, is available for consultation, or the librarian will mail copies of references to articles on any subject desired.

While it is not so reported in the booklet, the association has already been able to render distinguished service in a great number of instances. It has supplied references on all subjects submitted. It has provided answers on definite questions. In a number of cases, it has compiled complete bibliographies of selected subjects. Some of these are air conditioning for process rooms; ventilation of homes; solar radiation; dust pollution of the atmosphere in large cities; pitot tubes used in measurement of air velocities; gas refrigeration; gas heated radiators; fuel economy and comparative study of all kinds of fuel; hot water heating; radiators and radiator testing; manufacture and use of wood briquettes for domestic heating; welding, with special reference to pipe welding; and an interesting bibliography was prepared for Professor Philip Drinker of the *Harvard School of Public Health* to be used as a part of the international bibliography on psycho-physiological effects of the worker's physical environment. This bibliography is a part of a world-wide study inaugurated during the last conference on psycho-technology held in Utrecht in September, 1928.

The *Association for Correlating Thermal Research* keeps in close contact with the research activities of various universities and private laboratories and in many instances has been able to render valuable service. It deserves the co-operation of everyone in the field.

Air and Accidents

A plant safety director, presenting the results of an investigation before a recent safety conference, reached the conclusion that fifty per cent of our industrial accidents may be attributed to bad "plant housekeeping" methods.

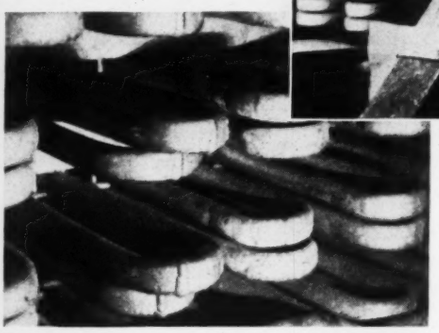
There are many factors that may be placed under the heading of plant housekeeping. One of the most obvious is the maintenance of proper atmospheric conditions. Accidents invariably are at a peak when employes are at the point of greatest fatigue—air conditioning will do much to reduce fatigue, and, consequently, costly and inexcusable accidents.

Many executives realize what air conditioning can do in a positive way for production and for increasing efficiency of workers, reducing labor turn-over and improving the morale of an industrial organization. Few have thought of it as a part of the safety program—as a means to be seriously considered when the prevention of accidents is sought. It is as important, if not more so, as proper safety guards on machinery, adequate and correct illumination, and the education of employes in the principles of safety.

Welding Galvanized Iron Pipe

GALVANIZED iron pipe is used extensively for service conditions where uncoated iron or steel would be subject to corrosion. The coating of zinc which forms the galvanizing is applied by immersing the iron base material, whether sheet, pipe or other form, in molten zinc. A layer of flux on the surface of the zinc bath cleans the iron as it is introduced and insures a firmly adherent coating of zinc on the galvanized material. This process is known as hot-dip galvanizing.

Zinc has a relatively low melting point and also has the property of volatilizing when heated. Consequently, when galvanized material is heated the zinc coating does not melt and run on the surface, as might be expected, but instead the zinc vaporizes and burns, forming the familiar white fumes of zinc oxide. The welding of galvanized material has very often resulted in the loss of a certain amount of zinc coating from the surface immediately adjacent to the weld.



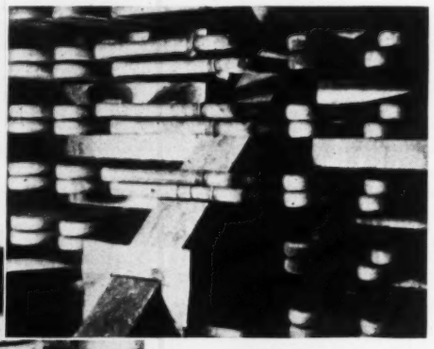


FIG. 1 — (ABOVE) FIELD WELDS

FIG. 2 — (LEFT) FIELD WELDS AFTER PAINTING

Some welded products such as coils can be completely welded prior to the galvanizing operation. In such cases, of course, the problem of welding galvanized material does not enter, as all of the welds are made in steel before coating with zinc.

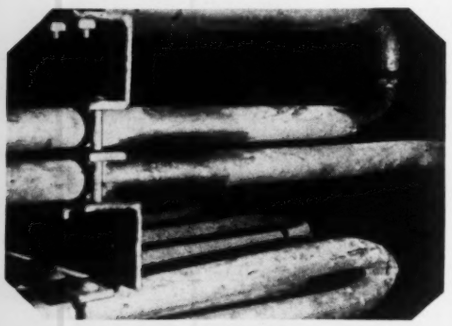


FIG. 3—FIELD WELD ABOVE, SHOP WELD BELOW

For the welding of galvanized pipe, either steel or bronze welding rod may be used.

Welding Galvanized Iron with Steel Welding Rod

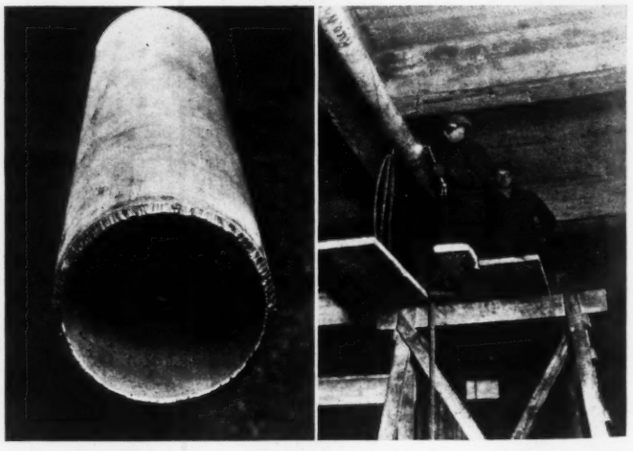
One large manufacturer of carbon dioxide refrigeration systems has standardized on the use of steel welding rod for making field welds in galvanized coils used in air conditioning systems. Many installations have been made in office buildings and theaters. The coils are fabricated in sections at the shop, the shop welds being made before the section is galvanized. During installation the galvanized sections are welded together. Fig. 1 shows a group of the field welds just after completion. The welds are later finished with a coat of aluminum paint, Fig. 2. Shop weld and field weld are contrasted in Fig. 3, the more abrupt bend shown in this illustration being the shop made one.

It is stated that no corrosion difficulties have been experienced due to loss of zinc coating from the inside of the pipe during the process of welding.

Bronze-Welding Galvanized Iron Pipe

Bronze-welding has been found very effective for the installation of galvanized piping for water and other industrial services. The installation is made in exactly the same manner as a steel line. Pipe can be cut and beveled with the cutting blowpipe without serious injury to the galvanized coating, Fig. 4. It is then lined up, tack-welded and welded in accordance with the usual procedure for bronze welding. The completed weld presents a very attractive appearance and in many cases no coating of paint or other protective material need be applied. Fig. 5 shows the installation of a bronze-welded 6-in. galvanized water line.

FIG. 4 — (LEFT) GALVANIZED PIPE BEVELLED WITH BLOWPIPE. FIG. 5—(RIGHT) BRONZE-WELDING GALVANIZED WATER LINE



Welding on longitudinal seams should be started on the outside at the end where the edges are tack-welded. Each seam should be completed without interruption, wherever possible. Two welds on two separate courses can be kept going continuously by having three welders.

PRACTICAL PIPING PROBLEMS

Valves Require Positive Indicators for Proper Operation— How to Make Them

by **W. H. WILSON**

Valve Indicators

The outside screw and yoke type of gate valve, with the rising stem, gives a positive indication that the valve is open, partly open or closed. This type of valve is appreciated by operating and maintenance men where there are a number of valves, as on branch lines, cross-connections and other piping. Boiler rooms, oil refineries, process piping, heating systems, waterworks, gas distribution lines and various other systems require valves with positive indicators, to insure safe and proper operation.

Installation requirements sometimes make it necessary to place these valves in locations not readily accessible, or points where the valve stem is out of the range of the operator's vision.

The following illustrations show how, in some cases, an indicator can be provided, at a reasonable cost.

Gate valves on pipe lines exposed to the weather are subject to corrosion of the stems or the accumulation of dirt in the threaded part of the stem. Unless the valves are operated frequently, the threads have to be cleaned when the valves are to be closed. This is also true of valves for underground water lines, when rising stem valves are used and are located in valve pits.

Fig. 1 shows an indicator that can be made to take care of this condition and it also provides a weather cap. A steel plate, with a hole in the center to fit over the valve stem, is fastened to the spokes of the valve hand wheel. U bolts and nuts are suitable for attaching this plate. A piece of wrought pipe that will slip over the stem, with a reasonable amount of clearance (large valves usually require 2-in. or 2½-in. pipe), is welded to this plate. The length of this pipe should be an inch or so longer than the distance that the stem projects above the hand wheel, when the valve is in the wide open position.

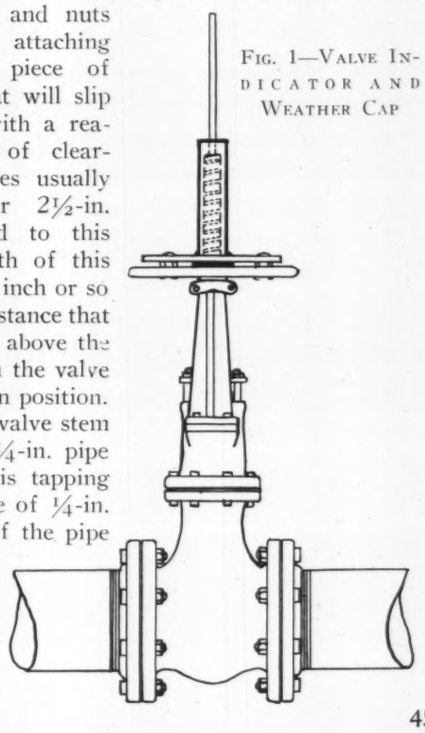


FIG. 1—VALVE INDICATOR AND WEATHER CAP

The top of the valve stem is tapped for a ¼-in. pipe thread and in this tapping is screwed a piece of ¼-in. pipe. The end of the pipe sleeve is closed by welding a round steel plate over the end. A hole is drilled in the center of this plate to permit the small pipe to pass through. This small pipe is as long as the distance from the top of the sleeve to the top of the valve stem, when the valve is in the closed position.

When the valve is closed, the small pipe will not be visible. When the valve is being opened, the small pipe will rise above the end of the pipe sleeve and act as a visible indicator as to how far the valve is open. Before assembling the pipe sleeve, grease should be applied to the threads of the valve stem, to assist easy operation of the valve.

Fig. 2 shows an addition to the above method which can be used to an advantage when the valve is located in a tunnel, below the floor line or other places that are not readily accessible. This device is similar to that shown in Fig. 1. The pipe sleeve is extended to the point where it is convenient to operate the valve. The end of the pipe sleeve, by a little forging work, is drawn down to fit the opening in the hand wheel (in some cases the opening in the hand wheel will have to be enlarged). The hand wheel is then securely fastened to the pipe sleeve.

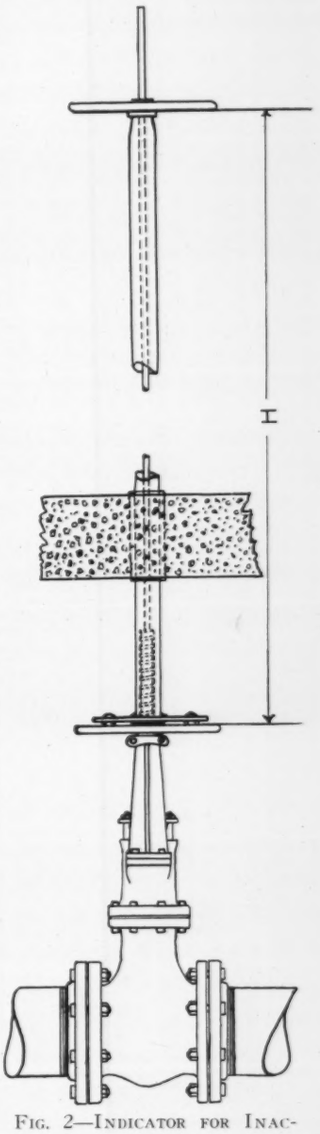


FIG. 2—INDICATOR FOR INACCESSIBLE VALVE

In case the dimension *H* is of any great length, suitable supports and guides must be provided and the lengths of pipe in the sleeve should either be welded together or, if threaded coupling joints are used, the couplings should have pins in the threaded joint to prevent the sleeve joints from becoming unscrewed when the valve is being opened.

The small pipe is screwed in a tapping in the end of the valve stem and extended through the sleeve similar to Fig. 1 and acts as an indicator, as previously described.